THE

GAS TURBINE

PROGRESS IN THE DESIGN AND CONSTRUCTION OF TURBINES OPERATED BY GASES OF COMBUSTION

BY

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PREFACE

THIS volume is intended to place in the hands of engineers and experimenters such theoretical and practical data as are now available in the solution of the problem of the gas turbine.

At the present time such machines are yet in the experimental stage, and it is still uncertain to what extent they may become generally practicable. There has, however, been expended much study and effort, both in the investigation of the theoretical principles upon which the gas turbine depends, and in the construction of machines intended to realize, more or less effectively, the possibilities which have been indicated by such studies.

Much of the information contained in this book is included in the transactions of learned societies, in the pages of periodicals, and in the records of private experimenters, and it is believed that by gathering in one volume the results of the work of English, French, German, and other organizations, the engineers and mechanics who are investigating the subject may be assisted by perceiving what has already been accomplished, and thus avoid unnecessary repetition of work which is already on record.

The gas turbine need not be a machine of exceedingly high thermal efficiency in order to be available for many purposes. The advantages accompanying a continuous turning effort, instead of the intermittent impulses of the reciprocating gas or gasoline motor, may, in many instances, overbalance a somewhat lower fuel economy; while the reduction in weight, consequent upon the attainment of a very high rotative speed, may become of controlling importance. It is therefore of the utmost desirability that all the conditions be taken into account, and it is for this purpose that the present volume has been prepared, collecting together the relative influence of the various elements of which the problem is composed.

The author desires to acknowledge the assistance which has been freely rendered to him in the preparation of this volume. To the memory of his colleague in the *Société des Ingénieurs Civils de France*, M. Réné Armengaud, he records his obligations for personal communications describing the experimental work conducted in the laboratory at St. Denis; and to M. Alfred Barbezat he desires to express his appreciation for the continuation of this most important work. To M. Armand de Dax, *Secrétaire Administratif* of the *Société des Ingénieurs Civils de France*, and to M. L. Sekutowicz, his colleague in the *Société*, he acknowledges the kind permission to translate the important paper of the latter author; and to Mr. Edgar Worthington, Secretary of the Institution of Mechanical Engineers (London) as well as to the author, Mr. R. M. Neilson, he is indebted for permission to reproduce the paper of the latter, as well as the discussion which it elicited. The writer also wishes to express his indebtedness to Dr. Sanford A. Moss, Dr. Charles E. Lucke, Prof. Sidney A. Reeve, Prof. Lionel S. Marks, and Dr. H. N. Davis, for valued suggestions and assistance.

HENRY HARRISON SUPLEE.

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INTRODUCTION.

ALTHOUGH the gas turbine was one of the earliest forms of combustion motor it failed to attain practical or commercial importance, and it is only since the steam turbine has reached its present commanding position that the possibility of developing the gas turbine in similar manner has been seriously considered.

The practical difficulties in the way of the realization of a successful gas turbine are very great. The high temperatures involved demand especial care in order that the strength of the material may not be unduly affected. The high rotative speeds required, if high efficiencies are to be secured, render the mechanical problems connected with centrifugal action more serious even than with the steam turbine; while the doubt as to the action of hot gases in diverging nozzles renders an important element in the theory yet uncertain.

At the same time there has been going on, during the past few years, in Europe and in America, some very effective experimental work upon the gas turbine; while the theory has also been made the subject of elaborate study by English, French, German, and American scientists.

Most of the information regarding this work is in a form unavailable for the practical engineer and investigator. The theoretical discussions are, for the most part, contained in the transactions of professional societies; much of it in languages other than English. The practical experiments are being conducted behind closed doors and reliable information is not generally attainable in detailed form.

It has therefore been thought desirable to gather under one cover the most important papers which have appeared upon the subject of the gas turbine in England, France, Germany, and Switzerland, together with some account of the work in America, and to add to this such information upon actual experimental machines as can be secured.

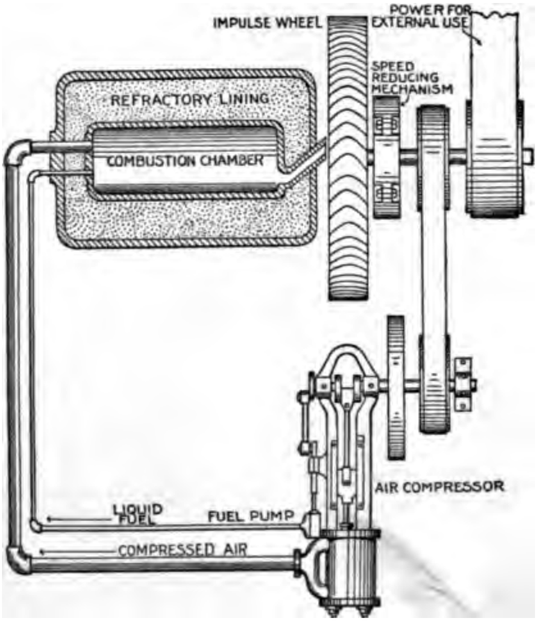

FIG. 1.—Scheme of gas turbine with reciprocating compressor.

In the present state of the art this is all that can be done, but it is believed that this will aid materially in the conduct of subsequent work, and place in the hands of the gas-power engineer a collection of material not generally accessible or available in convenient form.

The general lines along which the plans of the various gas turbines, now under experimental investigation, are constructed, will be understood from the accompanying schematic diagram. This is substantially as given by Dr. Sanford A. Moss in connection with his thesis upon the gas turbine presented to the faculty of Cornell University in 1903.

The fuel, in this case some form of liquid hydrocarbon, is forced into a combustion chamber, together with the proper amount of compressed air for its combustion. The products of combustion are discharged through a diverg-

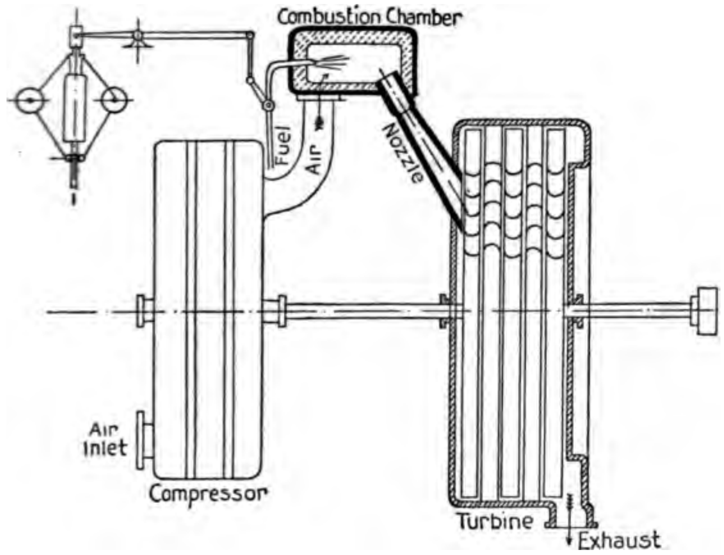

FIG. 2.—Scheme of gas turbine with multiple wheels and rotary compressor.

ing nozzle upon the buckets of an impulse wheel which is thus caused to rotate, a portion of the power developed being used to drive the compressor, and the remainder being available for external use.

Since one of the presumed advantages of the gas turbine over the ordinary gas engine is the substitution of continuous rotary motion for the reciprocating action of a piston in a cylinder, it is undoubtedly desirable that a rotary compressor be used, so that the reciprocating pump may be dispensed with. The general arrangement of such an apparatus, including also a multistage turbine, is here given, as devised by Mr. Rudolf Barkow (Fig. 2).

Here the air is drawn in and compressed by a rotary compressor, mounted on a continuation of the turbine shaft. The compressed air is delivered to the combustion chamber, into which the liquid fuel is also injected, and the combustion takes place under pressure, the gases and products of combustion passing through the diverging nozzle to the buckets and guide vanes of the turbine.

The extent to which these schematic forms have been developed from the earliest beginnings, and the lines along which theory and experiment have been pushed, will be seen in the following pages.

CHAPTER I.

HISTORICAL.

The use of the expansive action of heat upon elastic gases to operate a revolving wheel for the production of power is by no means recent; in fact it antedates the employment of a piston reciprocating in a cylinder for the same purpose. Considered in this broad sense there is no doubt that the windmill is entitled to be called a gas turbine, since the pressure of the moving air upon its sails can be traced to the currents produced by changes of temperature in the atmosphere.

Leaving aside the windmill, however, there can be little question as to the claims of the mediæval "Smokejack" to be considered as a gas turbine. This machine, the origin of which it is impossible to trace, has been attributed to Leonardo da Vinci and illustrations of it are to be found in his engineering sketch books. A somewhat later form is shown in the illustration, this being taken from an engraving in Bishop Wilkins's book "Mathematical Magick," published in 1648, the present engraving and following description being found in the edition of 1680.

After referring to the action of windmills and to Eolipiles, the learned bishop continues: "But there is a better invention to this purpose mentioned in Cardan,* whereby a spit may be turned (without the help of weights) by the motion of the air that ascends the Chimney; and it may be useful for the roasting of many or great joynts: for as the fire must be increased according to the quantity of meat, so the force of the instrument will be augmented proportionably to the fire. In which contrivance there are these conveniences above the Jacks of ordinary use:

* *Cardan. De Variet.* Rerum. *l.* 12, *c.* 58.

"1. It makes little or no noise in the motion.

"2. It needs no winding up, but will constantly move of itself, while there is any fire to rarifie the air.

"3. It is much cheaper than the other instruments that are commonly used to this purpose. There being required with it only a pair of sails, which must be placed in that part of the Chimney where it begins to be straightened, and one wheel, to the axis of which the spit line must be fastened, according to the following Diagram.

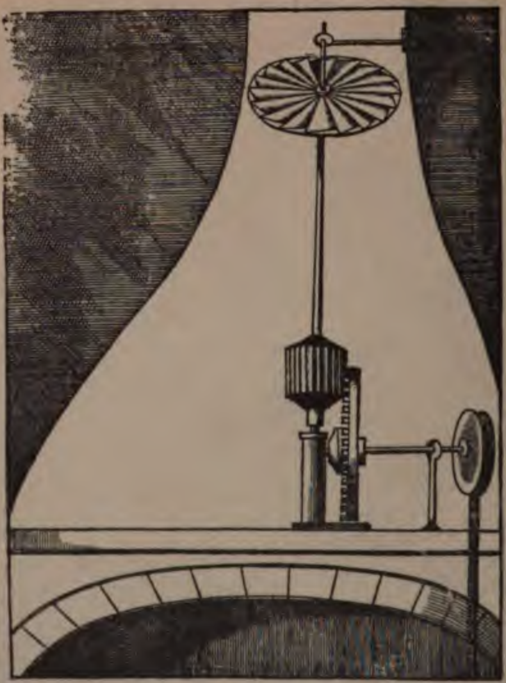

FIG. 3.—The smoke jack. The first gas turbine. From Bishop Wilkins's "Mathematical Magick," 1680.

"The motion to these sails may likewise be serviceable for sundry other purposes, besides the turning of a spit, for the chiming of bells or other musical devices; and there cannot be any more pleasant contrivance for continual and cheap music. It may be useful also for the reeling of yarn, the rocking of a cradle with divers the like demestick occasions. For (as was said before) any constant motion being given, it is easie for an ingenious artificer to apply it unto various services.

"These sails will always move both day and night, if there is but any fire under them, and sometimes though there be none. For if the air without be much colder than that within the room, then must this which is more warm and rarefied, naturally ascend through the chimney, to give place unto the more condensed and heavy, which does usually blow in at every chink or cranny, as experience shews."

After the smoke jack, the next proposition for a gas turbine appears to be that of Barber, who took out a British patent in 1791, No. 1833, which seems like a very complete anticipation of nearly all the most recent developments in this line. Barber's patent includes the distillation of the gas from wood, coal, or oil, its delivery, with the proper amount of air into a combustion chamber, and the discharge of the products of combustion upon the buckets of a turbine wheel.

He even went so far as to inject water into the combustion chambers to reduce the temperature, the mixed steam and gases acting upon the wheel.

The illustration, Fig. 4, gives an idea of Barber's patent.

The vessels marked 1, 1, are retorts for the production of the gas to be used, these being intended for the distillation of coal, wood, etc., by means of an external flame. When it is remembered that Murdock did not begin his experimental investigations into the manufacture of coal gas until 1792, one year after Barber's patent, and only made his results public in 1797, it will be seen that Barber was distinctly in advance of his time. The retorts shown in Barber's drawing are in duplicate, for alternate charging

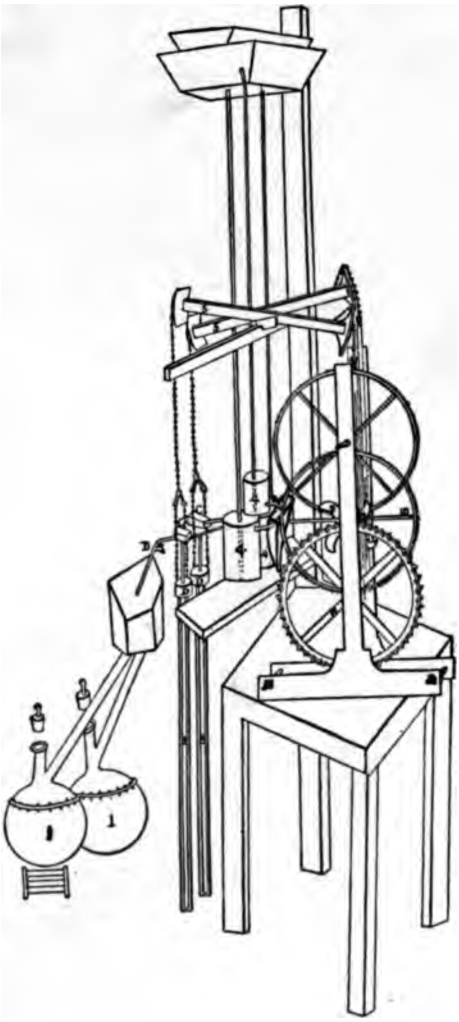

FIG. 4.—Barber's gas turbine, 1791.

and discharging, the gas being delivered into a cooling chamber *B*, from which it is drawn by one of the compressing pumps *C*, *D*, and delivered to the receiver 4, from which it passes to the triangular-shaped combustion chambers. The other compressing pump delivers air and vapor of water into the combustion chamber, and the products of combustion are discharged upon the buckets of the wheel to effect its rotation. The drawing shows the reducing gearing for operation of the compression pumps, the power to be taken from the upper gear shaft.

It is evident that Barber's machine involved constructive problems altogether unsolved in his time, but the apparatus was surprisingly complete in its conception, including combustion at constant pressure, with pumps for the supply of air and fuel, together with the use of vapor of water for the reduction of temperature, and a train of gear wheels for the reduction of the speed.

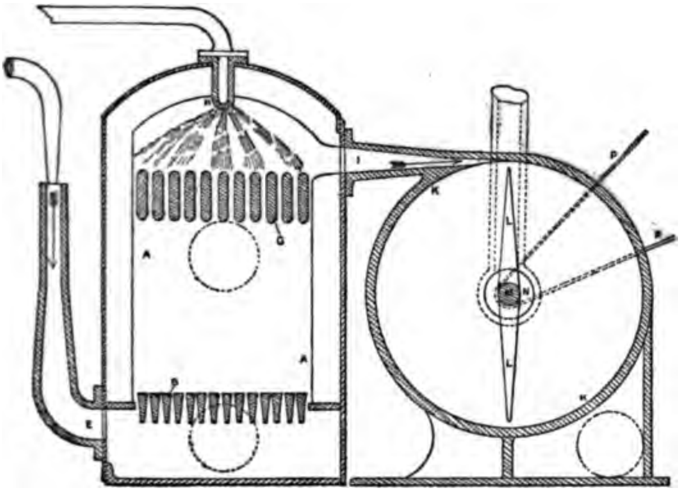

Fig. 5.—Fernihough's turbine, 1850.

Nothing seems to have been done for more than fifty years after the patent of Barber, but in 1850 a mixed steam and gas turbine was proposed by W. F. Fernihough, and patented in Great Britain, No. 1328, of 1850. This apparatus, Fig. 5, consisted of a chamber *A*, lined with refractory material, and fitted with a grate *B*, on which the fuel was ignited. Air was supplied under pressure through *E*, while water was sprayed from above at *H*, and the mixture of steam and the gases of combustion were delivered through the nozzle *I* upon the wheel *L*, *L*.

In the mean time Burdin, in France, had proposed, in 1847, to make a hot-air turbine, using a multiple-wheel rotary compressor to deliver air through a heating chamber to a corresponding rotary motor. This plan was included in the remarkable communication of Tournaire, presented to the *Académie des Sciences* in 1853. The original memoir of Tournaire is remarkable in many ways, both for the breadth of its conception of the problem, and also because it refers to "elastic fluid turbines," not limiting the action to steam, but including hot air and gases, and thus distinctly including the gas turbine. In view of the importance of this communication it is here translated entire, from the *Compte Rendu des Séances de l'Académie des Sciences* of March 28, 1853, pp. 588-593.

"APPLIED MECHANICS.—Note upon multiple and successive-reaction turbine devices for the utilization of the motive power developed by elastic fluids; by M. Tournaire, *Ingénieur des Mines*. Commission: MM. Poncelet, Lamé, Morin, Combès, Séguier:

"Numerous attempts have been made to cause the vapor of water or other gaseous substances to act by reaction upon the blades or passages of rotative apparatus similar to turbines or other hydraulic wheels; but down to the present time these inventions have not been crowned with practical success. The economical application of the principle of reaction to machines operated by elastic fluids would nevertheless be of a very high degree of interest, since the moving portions would thereby be reduced to very small dimensions, and, in the great majority of cases, the transmission of the motion would be lightened and simplified. In a word, such machines would enable the same advantages to be realized as are found with hydraulic turbines compared with water-wheels of large diameter.

"Elastic fluids acquire enormous velocities, even under the influence of comparatively low pressures. In order to utilize these pressures advantageously upon simple wheels analogous to hydraulic turbines, it would be necessary to permit a rotative motion of extraordinary rapidity, and to use extremely minute orifices, even for a large expenditure of fluid. These difficulties may be avoided by causing the steam or gas to lose its pressure, either in a gradual and continuous manner, or by successive fractions, making it react several times upon the blades of conveniently arranged turbines.

"We must attribute the origin of the researches which we have made upon this subject to the communications which M. Burdin, *Ingénieur en Chef des Mines*, and *membre Correspondant de l'Institut*, has had the courtesy to make to us, and which go back to the close of 1847. M. Burdin, who was then engaged upon a machine operated by hot air, desired to discharge the compressed and heated fluid upon a series of turbines fixed upon the same axis. Each one of these wheels was placed in a closed chamber, the air to be delivered through injector nozzles and discharged at a very low velocity. The author proposed to compress the cold air by means of a series of blowers arranged in a similar manner. This idea of employing a number of successive turbines in order to utilize the tension of the fluid a number of times seemed to us a simple and fertile one; we perceived in it the means of applying the principle of reaction to steam and air engines.

"Since the differences in pressure, as used in steam engines, are considerable, it became evident that a large number of turbines would be required to give a sufficient reduction in the velocity of the fluid jet. The lightness and small dimensions of the moving parts permits of very high rotative speeds compared with those of ordinary engines.

"Notwithstanding the multiplicity of parts, it is essential that the apparatus should be simple in its action, susceptible of a high degree of precision, and that adjustments and repairs should be readily made. We believe that we have fulfilled these essential conditions by means of the following arrangements:

"A machine is composed of several independent motor axes, connected by means of pinions to a single wheel for the transmission of the motion. Each of these axes carries several turbines; these receive and discharge the fluid at the same distance from the axis.

"Between two turbines is placed a fixed ring of guide blades. The guides receive the discharge from one reaction wheel and give to it a direction and velocity suitable to act upon the following wheel. Each of these systems of fixed and moving organs is to be enclosed in a cylindrical case. The guide blades will form portions of rings or annular pieces placed in the fixed cylinder, and these should be fitted very exactly the one to the other. The turbines should also have the form of rings, and should be fitted to a sleeve attached to the shaft. Projections fitting into grooves secure the guides to the cylindrical case, and fasten the turbines to the shaft. The first set of guides, which act simply as injector nozzles, may be made in one solid piece, carrying the journal of the shaft. Nothing could be easier than to erect or dismount such an apparatus. In order to transmit the motion it is necessary that the shaft should pass through the end of the cylindrical case through an opening fitted with a tight packing; such a single stuffing box will answer for each series of reaction wheels.

"After having acted upon the turbines on the first shaft, and thus parted with more or less of its elasticity, the fluid is caused to act upon the turbines of the second series, and so on. For this purpose large openings connect the end of each case with the beginning of the one which follows.

These cases and passages may form portions of the same casting. Since the steam or gas expands in proportion to its passage through the blades of the turbines and the guides, it is necessary that these blades should offer passages of continually increasing size, and the last portions of the apparatus will have much greater dimensions than the first.

"As in the case of hydraulic reaction wheels, the last turbine on each shaft should discharge the fluid with a very low velocity. At its flow from the other turbines the fluid should have a velocity best adapted to its entrance into the passages between the guide blades. The motive power developed by these wheels will be produced, in great part, not by the extinction of the actual velocity of the fluid, but from the differences in pressures in entering and leaving the blades. This difference in pressures will produce a great excess in the relative velocity of discharge over the relative velocity of entrance, and, in order that this effect may be obtained, it will suffice, by reason of the continuity of the motion, for the orifices of discharge of the passages to be of smaller area than the entrance orifices; this corresponds, in general, to the arrangement in most hydraulic turbines. Considered with regard to the relative velocity of rotation, the velocity of flow through the passages of our turbines will be much greater than in the passages in ordinary reaction wheels, and, in consequence, they will be capable of utilizing a much greater proportion of motive power.

"As is the case in all kinds of machinery, there are many causes tending to diminish the useful effect of our apparatus, and to render it lower than the theoretical effect.

"One portion of the fluid will escape through the clearance intervals which must be left between the fixed and moving portions, and will have no effect upon the turbines and will not be directed by the guide blades. There will be produced shocks and eddies at the entrance and discharge of the buckets. The considerable friction, due to the narrowness of the passages, will absorb a considerable portion of the theoretical work.

"All these injurious effects are produced in hydraulic turbines, some with an intensity almost equal in degree, others, such as the frictional resistances, to a much less extent. These reaction wheels are, nevertheless, excellent machines. In order that our steam or hot-air machines should equal them in respect to the effective power utilized, a very perfect construction will be necessary, which it will perhaps be difficult to attain, because of the small size of the parts. But if we consider the results obtained with piston engines operated by steam we see that we may make a large allowance for losses before our turbines fail to give equally good results. Many of the causes of loss inherent in the use of pistons and cylinders will be avoided. Thus, the cooling effect due to radiation from the exterior walls in contact with the surrounding medium will become negligible, since our cylindrical casings offer a very small mass and volume, traversed by a very large flow of heat.

"In order that the application of our principles may be successfully applied to engines operated by elastic fluids it is necessary that great care and a very high degree of precision be given to the construction and erection of the parts, and that the dimensions and curves of the blades be carefully studied.

"It is necessary that the teeth of the gear wheels, which are operated at very high speeds, should run with great smoothness, without shock or vibrations; the helicoidal system of gearing of White will probably be found desirable. The shafts should also be held by outside collars in order that the metallic stuffing boxes may not be subjected to heavy pressures. The journals will receive the pressure parallel to the axis; this, however, will be small, on account of the small dimensions of the turbines.

"As for the regulators of the flow of the fluid, their functions will be performed by two slides or valves, one placed in the pipe connecting the engine to the generator, and the other in the opening through which the exhaust is discharged into the atmosphere.

"The principal advantage offered by the motors which we propose lies in the extreme lightness and small size which they offer. This is a point upon which we believe it unnecessary to insist at length. The present engines are too heavy and cumbersome, and are yet incapable of application to many purposes which are still accomplished by the physical effort of man. Without doubt the realization of our projects would extend widely the domain of mechanical power.

"Applied to steam motors we believe that our multiple turbines would permit a reduction in the dimensions of the reservoirs or generators of the fluid; because, the consumption of the motive material being continuous, the ebullition will be effected very regularly in the boiler, and there will be much less danger of the entrainment of a large proportion of water. If hot air be substituted for steam, as we may hope from the beautiful and fertile experiments of Ericsson, our turbines will replace, very happily, the enormous cylinders and pistons used by the Swedish engineer to receive the action of the compressed air. It remains to be seen if similar rotative apparatus may not be usefully employed for the compression of cold air. In case of success a complete mechanical revolution will be effected not only with regard to the quantity of combustible consumed but also in the matter, not less important, of the masses and volumes which enter into machine construction."

It seems surprising that the clearly expressed ideas of Tournaire failed of immediate realization, especially as they were passed in review under the eyes of such a committee of mechanical specialists as Morin, Lamé, and Poncelet; but it is probable that constructive difficulties, the extent of which was fully realized by Tournaire, stood in the way. His work seemed to have been almost entirely overlooked until recently, but there is no doubt that he fully grasped the problem, as the text of his communication to the French Academy shows.

At the present time the term "elastic-fluid" turbine appears in nearly all patent specifications for such machines, their scope not being limited to steam alone. Tournaire not only used this very expression, but also foresaw the application of the multiple-turbine principle to pressure blowers as well.

He further saw that high fuel economy, while probably attainable with the turbine, was not the only advantage, but that material reduction in weight and in bulk might also be attained, points which to-day are of even more importance than they were fifty years ago.

An interesting forecast of the practicability of the gas turbine appears in the fifth edition of Bourne's large treatise on the steam engine, published in 1861. Discussing the advantages of superheated steam, Mr. Bourne says:

"Steam of a high temperature will, therefore, be more economical in its use than steam of a lower temperature, and surcharged steam being much hotter than common steam is consequently more advantageous. After all, however, the temperatures which it is possible to use with any kind of steam in an engine are too low to render any very important measure of economy possible by their instrumentality. We are, therefore, driven to consider the applicability of other agents, the most suitable of which appears to be air, and this brings us back to the point from whence we started at the commencement of the present chapter. Small measures of improvement are worth very little consideration when great and important steps of progress are apparently within our reach, and to us *it appears quite clear that the prod-*

ucts of combustion may be employed to produce motive power, not through the instrumentality of a cylinder and piston, but rather by means of a turbine or an instrument like a smoke jack or Barker's mill, and which may be made to work in water or some other liquid. In this way very high temperatures may be dealt with, and it is only by employing very high temperatures that any very great step of improvement is to be attained."

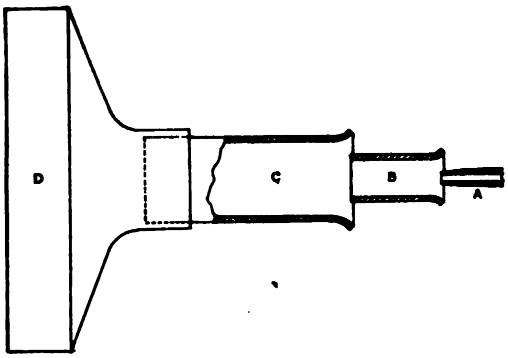

Fig. 6.—Boulton's multiple jet system

In 1864 the problem of combustion at constant pressure, in connection with the operation of a gas turbine, was investigated by M. P. W. Boulton, and his British patent, No. 1636 of 1864, contains some points of interest, in the light of what has been done since. He realized that the high velocity of the jet of gases issuing from the nozzle offered a practical difficulty, and proposed to remedy this by the use of successive induced jets of increasing volume and consequently lower velocity. This is shown in Fig. 4, the gases being delivered through the nozzle *A*, inducing a current in *B*, and this again in *C*. The turbine is represented at *D*, operated by the increased volume of fluid at the reduced velocity.

Another method proposed by Boulton for maintaining combustion at constant pressure is shown in Fig. 7. The gas is burned at *A*, in a chamber *C*, under water, the products of combustion passing up through the water between the baffle plates *E*, *E*, and the mixed gases and steam being delivered to the turbine from the top of the chamber *B*.

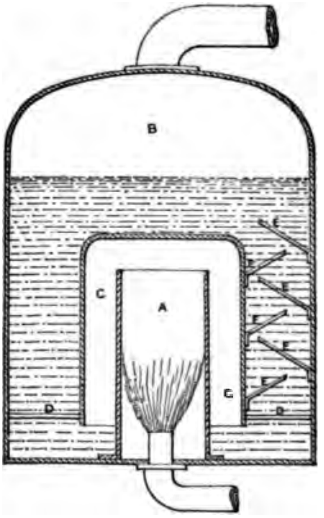

FIG. 7.—Boulton's constant-pressure combustion chamber.

The idea of combustion at constant pressure to furnish an elastic fluid composed of hot air and products of combustion for use in a turbine appears to have occupied the attention of a number of engineers from 1870 onward. John Bourne, the well-known British engineer and writer on the steam engine, took out two patents, one in 1869 and the other in 1870, relating to the combustion of coal dust for the production of gases for use in a turbine. His plans included the dilution of the gases with air and with the vapor of water, and involved the use of high pressures, up to 1,000 pounds per square inch. Bourne's patents refer entirely

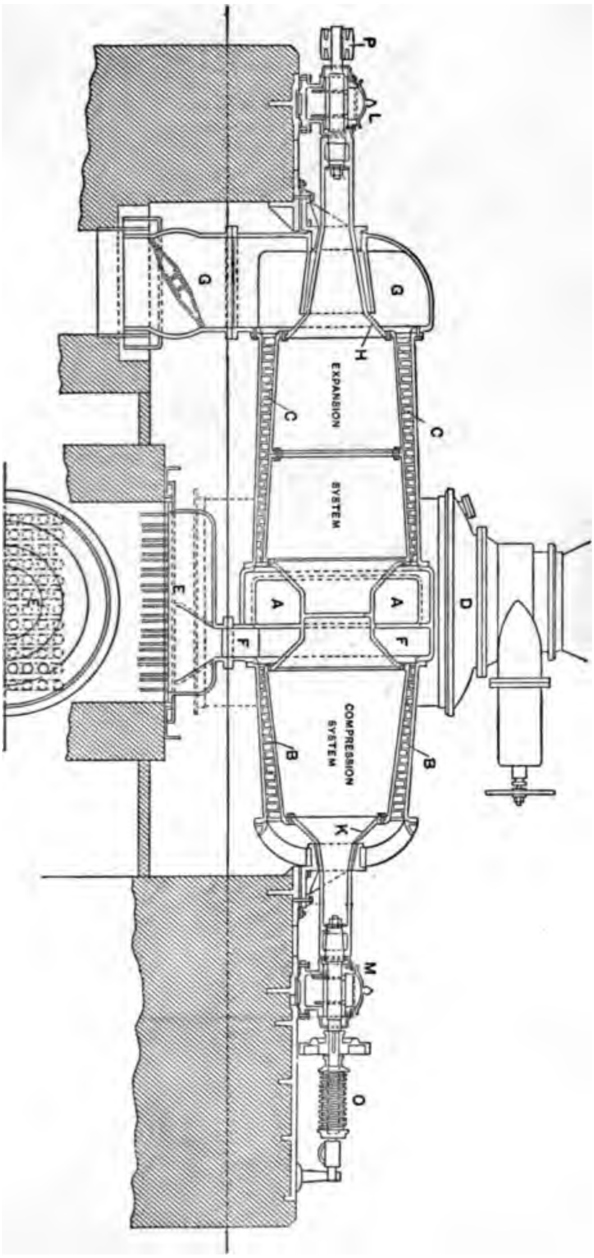

FIG. 8.—General arrangement of Stolze hot-air turbine.

to the production of the working fluid, and do not give any details of the turbine which he proposed to use.

Another British patent of about the same time is that of James Anderson, this including the combustion of a mixture of gas and air in the combination chamber or channel, the gases resulting from the combustion being led into a reaction turbine. He also proposed to make the combination chamber in the arms of the turbine itself.

It does not appear that any of these plans were ever put into actual operation. In 1872, however, we find that Dr. F. Stolze, of Charlottenburg, near Berlin, applied for a patent from the Prussian Government for a so-called "fire turbine," this practically being the same as the machine experimented upon by Burdin in 1847 and described by Tournaire in his communication to the French Academy of Sciences. The general scheme of the Stolze turbine is shown in Fig. 8, there being a multiple turbine compressor and a multiple power turbine on the same shaft, the compressed air being passed through a heating chamber and thus deriving energy from the heat of the fuel before passing to the power turbine. The exterior of the Stolze turbine is shown in Fig. 9, this representing his experimental machine at Charlottenburg.

The early work of the Hon. C. A. Parsons is generally supposed to have related wholly to the steam turbine, but in his original patent of 1884 (British Patent No. 6735) the following reference to the gas turbine occurs:

"Motors, according to my invention, are applicable to a variety of purposes, and if such an apparatus be driven, it becomes a pump and can be used for actuating a fluid column or producing pressure in a fluid. Such a fluid pressure-producer can be combined with a multiple motor, according to my invention, to obtain motive power from fuel or combustible gases of any kind. For this purpose I employ the pressure-producer to force air or combustible gases into a furnace into which there may or may not be introduced other fuel (liquid or solid). From the furnace the products of combustion can be led in a heated state to the multiple motor which they actuate. Conveniently, the pressure-producer and multiple motor can be mounted on the same shaft, the former to be driven by the latter; but I do not confine myself to this arrangement of parts. In some cases I employ water or other fluid to cool the blades, either by conduction of heat through their roots or by other suitable arrangement to effect their protection."

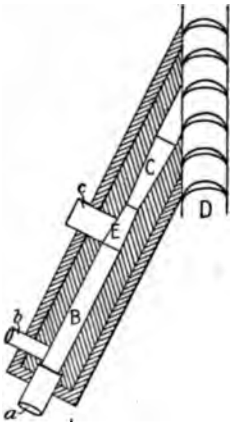

Fig. 10.—Combustion nozzles of De Laval gas turbine, 1893.

In 1893 De Laval proposed to deliver compressed air into a combustion chamber into which a liquid fuel was sprayed, the products of combustion being directed upon the blades of a wheel similar to that of the steam turbine known by his name. The general arrangement is shown in Fig. 10. The compressed air enters at *a* and the sprayed combustible at *b*, the combustion taking place in the space *B*. At *c* provision is made for an injection of water if necessary, the gaseous products passing through the nozzle *C* to the wheel *D*.

The first patent of M. Charles Lemale was taken out in 1901, followed in 1903 by a more complete development of the combustion chamber and expansion nozzle. M. Lemale, in conjunction with the late M. Réné Armengaud, continued to experiment with the gas turbine, under the auspices of the *Société des Turbomoteurs*, and the results of this work will be given hereafter at length.

In the United States Dr. Sanford A. Moss published, in 1903, a discussion of the subject of the gas turbine, in the form of a thesis presented to the faculty of Cornell University, this containing an examination of the thermodynamics of the gas turbine and a brief account of some experimental work.

The question has been discussed from a theoretical viewpoint by Mr. R. M. Neilson in a paper presented before the Institution of Mechanical Engineers in October, 1904, which with the discussion it evoked will be given in a following chapter.

It was also very fully examined by members of the *Société des Ingénieurs Civils de France* in consequence of an important paper by M. L. Sekutowicz, presented at the session of February 2, 1906, the discussion being taken up by MM. J. Deschamps, Réné Armengaud, Jean Rey, G. Hart, L. Letombe, and A. Bochet. M. Armengaud presented an important paper upon the subject before the Mechanical Section of the International Engineering Congress at Liége in June, 1905, this having been revised for publication in Cassier's Magazine for January, 1907.

In the *Schweizerische Bauzeitung* for August 27, 1904 there appeared an analysis of the action of the Armengaud and Lemale gas turbine by Alfred Barbezat, while two papers by Dr. Charles E. Lucke in the *Engineering Magazine* of April, 1905, and August, 1906, and one by Professor Sidney A. Reeve in the same magazine for June, 1905, formed current contributions to the theory of the subject.

An elaborate investigation of the practicability of the gas turbine was published in the *Zeitschrift für das Gesamte Turbinenwesen* by A. Baumann, of Zwickau, this appearing in the issues between December 15, 1905, and May 20, 1906. Several pamphlets upon the subject have appeared in Germany, among which may be mentioned: *Studien zur Frage der Gas-Turbine* (Studies upon the Question of the Gas Turbine), by Rudolf Barkow; *Ein Praktisch Brauchbare Gas-Turbine* (A Practical, Useful Gas Turbine), by Dr. Richard Wegener; and *Die Aussischten der Gas Turbine* (The Outlook for the Gas Turbine), by Felix Langen.

The most important work from a theoretical point of view is given in the discussions before the Institution of Mechanical Engineers, in London, and the Society of Civil Engineers of France, and these are given practically entire, followed by abstracts of other papers, and as much information concerning actual machines as can at present be made public.

CHAPTER II.

THE DISCUSSION BEFORE THE INSTITUTION OF MECHANICAL ENGINEERS.

On October 21, 1904, Mr. R. M. Neilson, Associate Member of the Institution of Mechanical Engineers read before the Institution at its house in London, a paper entitled: "A Scientific Investigation into the Possibilities of Gas Turbines." By the kind permission of the Council of the Institution this paper is here given entire, together with the discussion which it elicited from the membership, this forming one of the most important contributions to the question which has yet appeared in England.

In examining this paper and the discussion upon it, it must be remembered that at the time of its presentation, 1904, the investigations of Dr. Charles E. Lucke upon temperatures and pressures in free expansion of hot gases in nozzles had not yet been made public, nor had the work of Professor Rateau in the construction of turbine air compressors of high efficiency been completed.

A SCIENTIFIC INVESTIGATION INTO THE POSSIBILITIES OF GAS TURBINES

By Mr. R. M. Neilson
Associate Member, Institution of Mechanical Engineers.

A prophecy expressed frequently in engineering circles at the present day is that turbines actuated by hot gases other than steam will eventually come to the front as prime movers. The idea of employing hot gases (other than steam) to drive turbines is by no means new; but the success of the steam turbine has recently brought the question into prom-

Fig. 9.—The Stolze hot-air turbine.

inence. Although the subject is interesting and important, and although many minds seem to be considering it, there appears to be hardly any literature on the subject, except that which is found in patent records.

There is no doubt that many persons speak of the advantages of gas turbines without duly considering the difficulties to be encountered. There are probably many others who have valuable ideas on the subject, supported in some cases by experimental data, but who are apt to let their thoughts run in a groove and to consider (rightly or wrongly) that the only possible solution of the gas turbine problem lies in the particular direction in which they are working.

This Paper is written with the object of expressing and and comparing as concisely as possible the advantages and possibilities of gas turbines worked on different cycles, and the difficulties to be overcome to make these turbines a success. A further and more important object is to draw opinions from other engineers who have studied the question, and especially from those who have conducted experiments. If these objects be obtained, even in an imperfect manner, the author believes that a foundation of knowledge will be obtained and placed on record, which will be of considerable use to engineers who may be endeavoring or about to endeavor to produce practical machines.

Carnot's formula for the efficiency of an ideal heat engine

$$E = \frac{T_1 - T_2}{T_1}$$

is well known, but its real meaning is sometimes forgotten; and it may not be out of place here to put in a reminder that, in Carnot's cycle, *all* the heat is put in at temperature T_1 and *all* the heat withdrawn at temperature T_2. An increase in the range of temperature does not necessarily cause a thermodynamic gain, and it is possible largely to increase the range of temperature (as for example by superheating steam before use in a steam engine) without thermodynamically increasing the efficiency by more than a small percentage.

The greatest possible efficiency of a gas engine (reciprocating or turbine) working on Carnot's cycle between the limits of temperature 1600° C. (2912° F.) and 17° C., will be found to be:—

$$\frac{(1600+273)-(17+273)}{1600+273}=0{\cdot}85.$$

If the gas engine be an explosion motor with compression to 60 pounds per square inch above atmosphere, combustion at constant volume, and expansion to atmospheric pressure, the greatest possible efficiency between the same limits of temperature is only 0.50; and, in the engine work on the ordinary Otto cycle with the same compression and between the same limits of temperature, the greatest possible efficiency is only 0.37.

Efforts must therefore be made not so much to get the maximum and minimum temperatures respectively as high and as low as possible, but to get the mean temperature at which heat is given to the gas and the mean temperature at which heat is withdrawn from it respectively as high and as low as possible. Of these two temperatures the lower one is usually by far the more important. An ideal gas engine working on Carnot's cycle between the limits of temperature 2000° C. (3632° F.) absolute and 300° C. (572° F.) absolute will lose as much by an increase of 100° C. to the lower temperature as it will by a decrease of 500° C. from the higher temperature.

Coming now to discuss more particularly gas turbines, there are four cycles on which it seems to the author that these could be worked with the possibility of good results. Two of these are what Mr. Dugald Clerk designates Type 2 and Type 3.* The author will call them respectively Cycle I and Cycle II.

It has not been considered worth while to discuss the Carnot cycle at length, but a few remarks are made about it towards the end of the Paper (page 74).

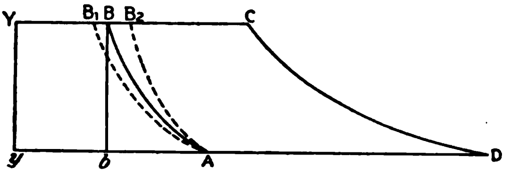

Fig. 11.—Pressure-volume diagram.

A pressure-volume diagram of an engine working on Cycle I is shown in Fig. 11, and an entropy-temperature diagram in Fig. 12.

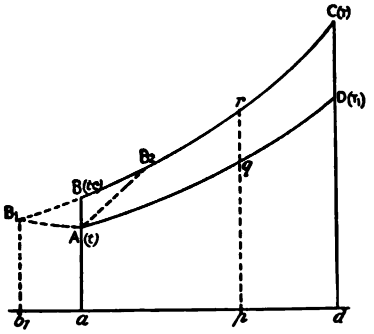

Fig. 12.—Entropy-temperature diagram.

The working fluid is compressed adiabatically from A to B. Heat is then supplied by combustion at constant pressure from B to C; the gas expands adiabatically from C to D, and the fluid is then cooled at constant pressure from D to A. Reciprocating gas engines have been worked on this cycle by Brayton and others, but have never come into common use. (The Diesel engine may be considered to belong

* "The Gas and Oil Engine," by Dugald Clerk (Longmans & Co.), Chapter III.

to this class, although no decided constant-pressure line is discernible on indicator diagrams taken from the engine.) One great difficulty that has been experienced in working reciprocating engines on this cycle is that of getting complete combustion during the period $B\ C$ without the charge occasionally firing back. If the air and fuel are brought into contact only on entering the cylinder, it is difficult to get good combustion during the period $B\,C$. If, on the other hand, the air and fuel are previously mixed together, it is difficult to prevent occasional firing back. Of course the chamber in which the air and fuel are mixed may be made strong enough to stand explosions; but any back firing upsets the regular working of the engine and is otherwise objectionable.

It has been proposed for gas turbines to cause air and fuel to unite in a nozzle, which thereafter diverges, the idea being that the air and fuel will combine on meeting each other, and the hot products of combustion will then acquire a high velocity in the divergent nozzle with which velocity they will enter the turbine buckets. The results of a trial of such a scheme would be interesting. The author doubts if the combustion would be quick enough to give a good efficiency. If, however a combustion chamber of ample size were provided in which the burning gases could rest a short interval before passing to the turbine, better results could, in the author's opinion, be expected. The air and fuel would be separately pumped into the chamber from which the products of combustion would flow continuously and uniformly by one or more passages into the turbine.

At any rate the difficulties should not be as great with turbines working on this cycle as with reciprocating engines, as the latter have to receive the hot gases intermittently, while the turbine receives a continuous flow. This is an important point as regards controlling the flame. With an engine of the Brayton type the fuel has to be ignited in the cylinder for every working stroke, and the supply of gas to the flame has to be cut off for every working stroke. With a turbine the fuel and air could be supplied at a constant velocity to the flame and a steady flame maintained without interruptions. This is important, because, if a mixture of air and fuel be always supplied to the flame with a velocity greater than the velocity or propagation of the flame, there can of course be no firing back, and this result can be obtained without the use of a wire-gauze screen. The maintaining of this velocity of supply to the flame above the required minimum when starting and stopping the motor, and when running at low powers, is of course a problem to be considered, and some consideration is given to it later on (pages 77 and 78). The strength of the mixture of air and fuel should be kept constant. The power of the turbine can be varied by other means, which will be referred to later (pages 77 and 78). It must be noted that if the air and fuel are compressed adiabatically to a sufficient extent, which depends on the nature of the fuel, combustion will occur immediately the two are brought into contact with each other. It is therefore necessary in such cases to keep the air and fuel apart until the instant when combustion is desired. It must also be noted that with a turbine there will be no hot waste gases mixed with the fresh air and gas to be compressed.

This cycle allows of a fairly high ideal efficiency being obtained with a moderate maximum temperature. Now a moderate maximum temperature is of the utmost importance in the case of a turbine of the Parsons type. A Parsons turbine with steel blades could probably be designed without any great difficulty to stand a temperature of about 700° C. (1292° F.) without any water jacketing or cooling devices of any sort (except for the bearings). With temperatures above this, the blades would need to be cooled. This would necessitate a radical alteration in design. The question of designing a turbine to stand high temperatures will be considered later on. It is only desired here to point out that great difficulties with a certain class of turbine are avoided by keeping the maximum temperature moderate. The cycle under consideration may therefore have great advantages for turbines.

It had better be stated here that the author has made several assumptions with regard to the working fluid or fluids. These assumptions are as follows:—

1. That the specific heats of gases dealt with are constant at all temperatures and pressures, and are as follows:—

   Specific heat at constant pressure or $Kp = 0.238$.

   Specific heat at constant volume or $Kv = 0.17$

2. That weight per cubic foot of gases dealt with = 0.0777 pounds at a pressure of 15 pounds per square inch absolute and a temperature of 17° C.

3. That $PV =$ a constant for all pressures and temperatures.

4. That $PV =$ a constant for isothermal expansion and compression at all temperatures and pressures.

5. That combustion produces no change of volume except that due to change of temperature.

Some of these assumptions will probably be appreciably inaccurate in certain cases; but it seemed advisable to sacrifice something for simplicity and uniformity. As regards the variability of the specific heats, it has been thought better to assume constancy until more knowledge on the subject has been obtained and a scale of change (if any) has been agreed upon.

Pressures have been reckoned in pounds per square inch, and temperatures have generally been reckoned on the Centigrade scale, although for convenience the corresponding readings on the Fahrenheit scale have also been given. The numbers on the diagrams representing pressure and temperature are all representative of absolute pressures in pounds per square inch, and temperatures on the absolute Centigrade scale.

Referring to Fig. 12 (page 31), the heat absorbed by the fluid is represented in this figure by the area $aBCd$, and the heat abstracted or discarded by the area $aADd$. The heat converted into work is represented by the area $ABCD$, and consequently, if E represents the ideal efficiency of an engine working on this cycle,

$$E = \frac{\text{area } ABCD}{\text{area } aBCd}.$$

Now, as it can be proved * that $\frac{AB}{aB} = \frac{DC}{dC} = \frac{qr}{pr}$ where pqr is any ordinate cutting the lines ad, AD, and BC, which are all constant-pressure lines,

therefore $$E = \frac{AB}{aB} = \frac{DC}{dC}. \qquad (1)$$

Let t represent the temperature before compression.

Let t_0 represent the temperature at the end of compression.

Let T represent the temperature at the end of combustion.

Let T_1 represent the temperature at the end of adiabatic expansion.

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* Since all vertical lines represent adiabatic expansion, therefore, by the laws of adiabatic expansion,

$$\frac{\text{temp. at } A}{\text{temp. at } B} = \left[\frac{\text{press. at } A}{\text{press. at } B}\right]^{\frac{\gamma-1}{\gamma}} \quad \text{where } \gamma = \frac{Kp}{Kv}$$

Similarly $$\frac{\text{temp. at } q}{\text{temp. at } r} = \left[\frac{\text{press. at } q}{\text{press. at } r}\right]^{\frac{\gamma-1}{\gamma}}$$

But press. at A = press. at q, since AqD is a constant-pressure line; and press. at B = press. at r, since BrC is a constant-pressure line,

therefore $$\frac{\text{temp. at } A}{\text{temp. at } B} = \frac{\text{temp. at } q}{\text{temp. at } r}$$

therefore $$\frac{AB}{aB} = \frac{DC}{dC} = \frac{qr}{pr}$$

Then, from equation (1) and referring to Fig. 12,

$$E=\frac{t_c-t}{t_c}=\frac{T-T_1}{T}.$$

This can be proved quite well without any entropy-temperature diagram.* The diagram, however, shows the efficiency better.

It is important to consider the amount of negative work done and the ratio of this to the total or gross work. The negative work is the work of compressing the gas and delivering it in its compressed state. It is true that with some engines there is no work of delivery. In a reciprocating gas engine in which the gas is compressed in the motor cylinder, the only negative work (ideally) is that of compressing the charge; and, even when a separate cylinder is used for the compression, the work of delivering might be avoided. With a turbine, however, the fluid cannot be compressed in the motor; and, whatever arrangement is adopted, the compressed fluid will have to be delivered after compression. The author has, therefore, considered it better in all cases to include in the negative work the amount required to deliver the compressed gas. The motor proper of course gets the benefit of this work.

In Fig. 11 (page 31) the work to compress the gas is represented by the area *AbB*, and the work to deliver it in compressed state by the area *yYBb*. The total negative work is therefore represented by the area *yYBA*. The gross work of the motor is represented by the area *yYCD*, of which the part *yYBb* represents the work done before expansion, and the part *bBCD* the work done during expansion. By deducting the negative work from the gross work the net work is obtained; this is represented by the area *ABCD*. This net work is the same as that represented on the entropy-temperature diagram, Fig. 12 (page 31), by the area *ABCD*.

* See "The Gas and Oil Engine," by Dugald Clerk, pages 46–48.

Cycle I, Case 1.

If the gas is required to be used in a Parsons turbine without cooling arrangements the maximum temperature must not exceed 700° C. (1292° F.). A case with this maximum temperature will now be considered:—

In all cases—

Let t and p represent respectively absolute temperature C. and absolute pressure pounds per square inch before compression.

Let t_c and p_c represent respectively absolute temperature C. and absolute pressure pounds per square inch after compression.

Let T and P represent respectively absolute temperature C. and absolute pressure pounds per square inch after combustion.

Let T_1 and P_1 represent respectively absolute temperature C. and absolute pressure pounds per square inch after expansion to atmospheric pressure.

Let v represent one cubic foot of the fluid at temperature t and pressure p.

Let v_c, V and V_1 represent the volume of the same at t_c, p_c; T, P, and T_1, P_1 respectively.

Suppose that in all cases $t = 17°$ C. (290° absolute C.) and the corresponding pressure = 15 pounds absolute. First by compressing to 42 pounds absolute: t_c will then be 389° absolute C. This compression is shown by the line AB on the pressure-volume diagram, Fig. 13 (page 38), and on the entropy-temperature diagram, Fig. 14.

Let heat be supplied and the gas expand at constant pressure along the line BC till the temperature is 973° absolute C. Let the gas expand adiabatically along the line CD till the pressure falls to 15 pounds absolute. The fluid is then exhausted into atmosphere, and as the new charge is taken at the same pressure and at temperature t, it can be assumed that the discharged gas is cooled at constant pressure and used over again. Both diagrams can therefore be completed by the constant-pressure line DA.

The heat absorbed by the fluid is represented by the area $aBCd$ in Fig. 14, and the heat rejected by the area $aADd$. The heat converted into work is represented by the area $ABCD$ and

$$E = \frac{\text{area } ABCD}{\text{area } aBCd} = \frac{t_c - t}{t_c} = \frac{389 - 290}{389} = \frac{99}{389} = 0.25$$

The negative work is represented in Fig. 13 by the area $yYBA$, the gross work by the area $yYCD$, and the net work by the area $ABCD$,—

therefore $$\frac{\text{negative work}}{\text{gross work}} = \frac{\text{area } yYBA}{\text{area } yYCD} = 0.4.$$

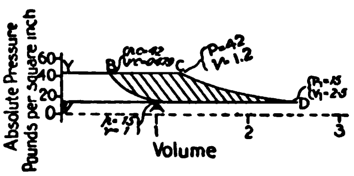


FIG. 13.—Cycle I, Case 1. Pressure-volume diagram.

The expansion line is carried right down to atmosphere. It should be possible in practice without difficulty to do this very nearly in a turbine, although the volume at D is 2½ times the volume at A. In dealing with large volumes and

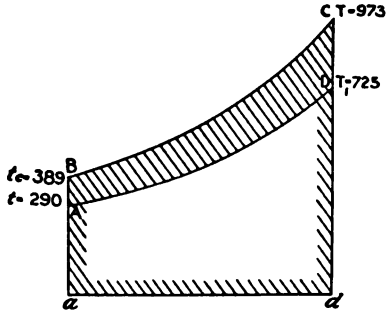


FIG. 14.—Cycle I, Case 1. Entropy-temperature diagram.

small pressures there is an immense difference between turbines and reciprocating engines. Reciprocating engines require large cylinders. These large cylinders, besides being objectionable on account of bulk and cost, necessitate great frictional losses. The low pressure dealt with is of little import as regards friction, which will be nearly the same whether the pressure is 13 pounds below atmosphere or 13 pounds above atmosphere. With a turbine, however, the large volume of the fluid does not necessitate such a bulky machine. Moreover in a turbine the friction depends on the pressure. With high pressures the friction is great, with low pressures very small. (In marine propulsion by steam turbines it is not considered worth while uncoupling the reversing turbines when the vessel is going ahead. These turbines are allowed to rotate (above their normal speed) in the low pressure which exists at the exhaust ends of the main low-pressure turbines.

Cycle 1, Case 2.

700° C. (1292° F.) must not, however, be considered as the limiting temperature for gas turbines. Much higher temperature can be employed if water-cooling or other cooling arrangements be used. Mr. Parsons has circulated steam for heating purposes through passages formed in the rings supporting the fixed blades of his radial-flow steam turbines.* Water could as easily be circulated, and there should be no great difficulty in passing the water also through the rings supporting the moving blades.

It has been proposed by Mr. Parsons and others to circulate water or other cooling fluid through the actual blades of a turbine, these being formed hollow. It has also been proposed to keep the blades of a single-wheel turbine cool by causing the actuating fluid to act only at one point of the circumference of the wheel, while a cooling fluid is projected onto the blades at another point.

By the employment of cooling devices a turbine might possibly be made to stand a temperature of 1500° C. (2732° F.) or even 2000° C. (3632° F.). 2000° C. is a very high temperature, and there would be great difficulty in devising and constructing cooling arrangements which would keep the blades in good working order when acted on

* "The Steam Turbine," by R. M. Neilson (Longmans and Co.), pp. 43–45.

by gas at a temperature approaching this. Let it be assumed, however, that 2000° C. is allowable for the maximum temperature; then, if the same compression is kept as in Case 1, the ideal pressure-volume and entropy-temperature diagrams will be as shown in Figs. 15 and 16. In these Figs. the line CD has been reproduced from Figs. 13 and 14 (page 38), and is shown in dotted lines in order that the two cases may be readily compared.

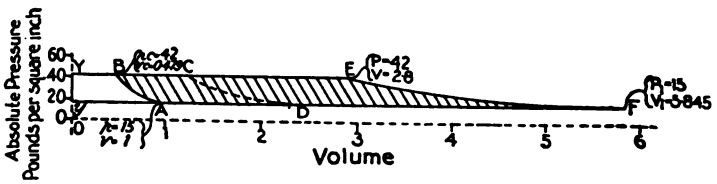


FIG. 15.—Cycle I, Case 2. Pressure-volume diagram.

Referring to Fig. 16 the heat absorbed by the fluid is represented by the area $aBEf$, the heat rejected by the area $aAFf$, and the heat converted into work by the area $ABEF$.

Therefore
$$E=\frac{\text{area } ABEF}{\text{area } aBEf}=\frac{t_c-t}{t_c}$$
$$=\frac{389-290}{389}=0.25.$$

The increase in the maximum temperature has, therefore, added nothing to the efficiency, and this will always be the case if the initial temperature and pressure are unchanged and compression is made to the same amount. That is to say, as long as the constant pressure lines are started from the same points, A and B, they can be extended any distance to the right and connected by any adiabatic line; E will remain unchanged. In Fig. 16 the additional area $dCEf$ is divided by the line DF in the same ratio as the original area $aBCd$ is divided by the line AD.

The negative work (in Case 2) is represented in Fig. 15 by the area $yYBA$; it is the same as in the last case. The gross work is represented by the area $yYEF$, and the net work by the area $ABEF$,

Therefore $$\frac{\text{negative work}}{\text{gross work}}=\frac{\text{area } yYBA}{\text{area } yYEF}=0.171.$$

The ratio of negative work to gross work has, therefore, been very considerably diminished.

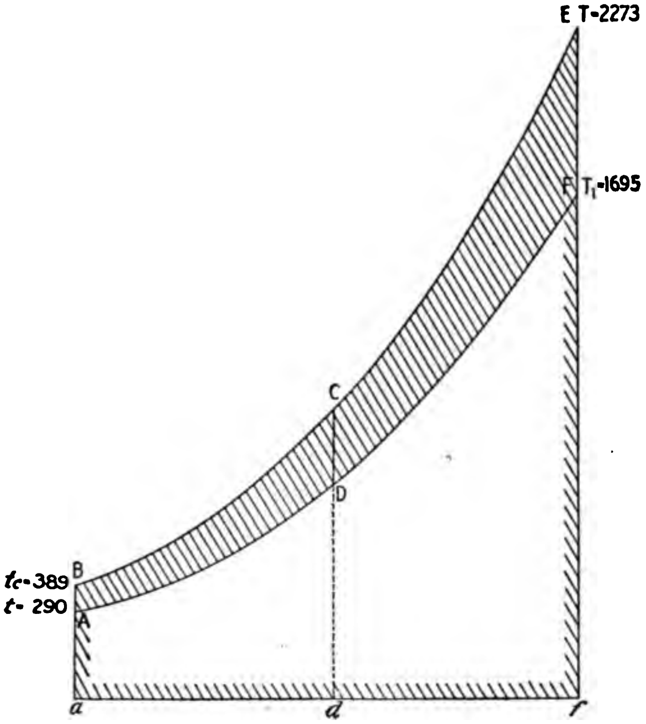


FIG. 16.—Cycle I, Case 2. Entropy-temperature diagram.

Cycle I, Case 3.

In Case 1 it was necessary to have a low compression because a high compression with a maximum temperature of only 700° C. (1292° F.) would have given an impractically high value to the ratio of negative work to gross work. In fact this ratio was high even with the low compression adopted.

With the maximum temperature raised to 2000° C. (3632° F.), however, a much higher compression can be adopted. Suppose a compression to 300 pounds per square inch absolute is adopted. This will make t_o 682.5° absolute C. (1260.5° F.). The pressure-volume and entropy-temperature diagrams will then be as shown in Figs. 17 and 18.

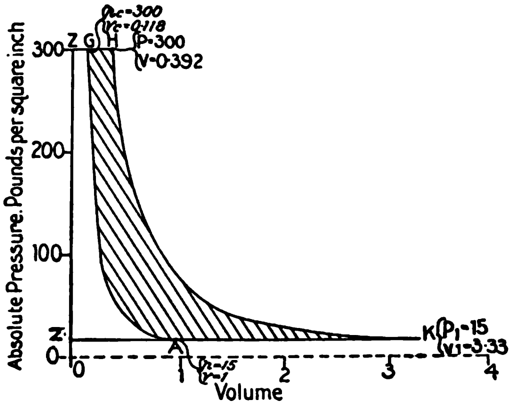


Fig. 17.—Cycle I, Case 3. Pressure-volume diagram.

Referring to Fig. 18 it is seen that

$$E = \frac{\text{area } AGHK}{\text{area } aGHk}$$

$$= \frac{t_o - t}{t_o} = \frac{682.5 - 290}{682.5}$$

$$= 0.58$$

which is much better than (more than double) that in Cases 1 and 2. There is, however, the inconvenience of a high compression, and compared with Case 1 more heat is likely to be lost through radiation owing to the higher average temperature. This question of radiation will be more or less important according to the type of turbine.

The negative work is represented in Fig. 17 by the area *zZGA*, the gross work by the area *zZHK*, and the net work by the area *AGHK*.

Therefore $$\frac{\text{negative work}}{\text{gross work}}=\frac{\text{area } zZGA}{zZHK}=0.3.$$

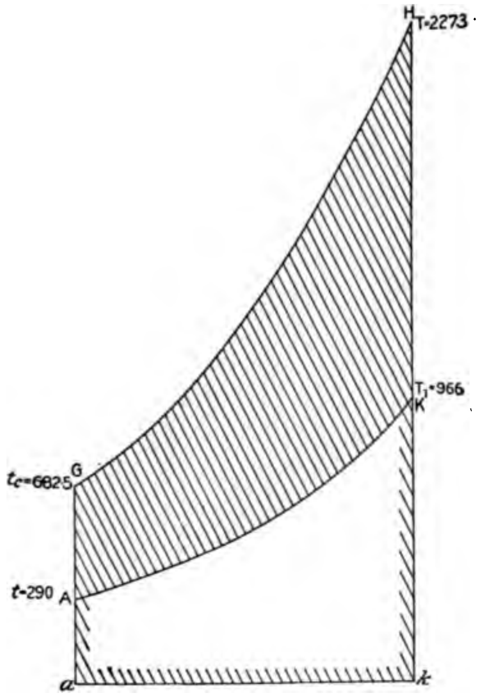


FIG. 18.—Cycle I, Case 3. Entropy-temperature diagram.

Cycle I, Case 4.

It will be interesting to find what efficiency can be obtained with a maximum temperature of 2000° C. (3632° F.) by increasing the compression till the ratio of negative work to gross work is 0.4—the same as in Case 1. This ratio will be attained when t_c = 909° absolute C. (1668° F.), which corresponds to a pressure of 818 pounds absolute.

Then $$E=\frac{909-290}{909}=0.68.$$

The pressure-volume and entropy-temperature diagrams for this case are given in Figs. 19 and 20.

The line *BC* is shown dotted on Fig. 20 to allow Case 4 to be compared with Case 1.

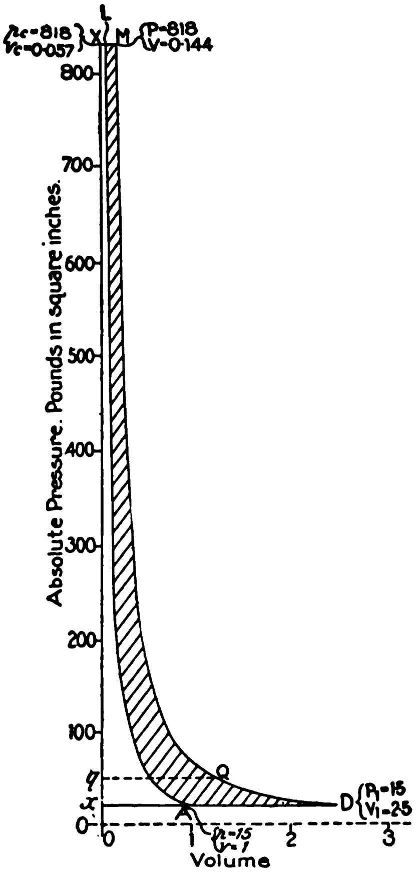


FIG. 19.—Cycle I, Case 4. Pressure-volume diagram.

The sharp corner at *M* would likely be rounded off in practice. This would reduce the efficiency slightly. It would also, however, reduce the maximum temperature, and for this reason it might be advantageous in some cases to round off the corner intentionally.

In every case it has been assumed that the compression is adiabatic; it is usually important that it should be at least nearly so. If, for example, in Figs. 11 and 12 (page 31) the compression, instead of being along the adiabatic AB had been along the line AB_1, which is below the adiabatic line, that is, if heat had been allowed to escape during the

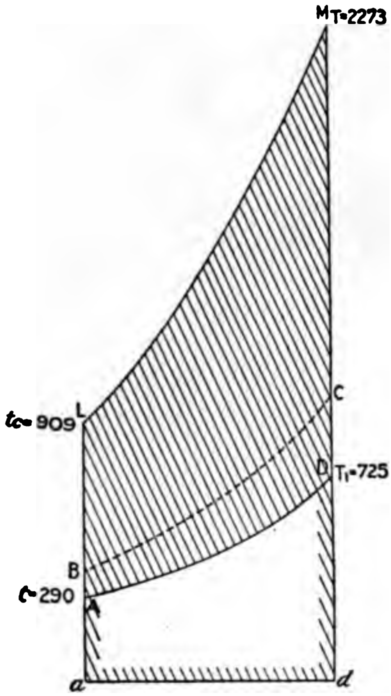


FIG. 20.—Cycle I, Case 4. Entropy-temperature diagram.

compression, the heat absorbed by the fluid for the same value of T would have been increased by the area b_1B_1Ba in Fig. 12, while the heat converted into work would have been increased only by the relatively small area AB_1B. E would, therefore, have been reduced.

If on the other hand the compression had been along the line AB_2, which is above the adiabatic, that is, if heat had been put into the fluid during compression, the heat absorbed and the heat converted into work would both have been reduced by the same amount, namely, the area ABB_2. E would, therefore, obviously be reduced in this case also, assuming that the heat put into the fluid during compression is obtained by the combustion of fuel.

If, however, the heat put into the fluid during compression is obtained for nothing—if, for example, it is heat that would otherwise be radiated away or carried away by convexion—the effect on E is not obvious.

A compression along the line AB_2, Figs. 11 and 12, will give a higher value to E than a compression along the line AB, if the heat absorbed during the compression AB_2 is got for nothing, and if the two cases are otherwise the same; but a compression along the line AB_2 produces a higher ratio of negative work to gross work. This will be clear from Fig. 11. Now with this ratio of negative work to gross work, a still higher efficiency could be obtained by keeping the compression adiabatic and continuing it further. A hot compression, such as along the line AB_2, when the heat is got for nothing, may be advantageous in a few cases, viz., T_1 is low compared with t_c; but generally such a compression will be harmful.

It is, in general, disadvantageous to heat the air or fuel before compression, no matter what be the source of heat.

If gas is allowed to enter a water-cooled turbine at a high temperature, such as 2000° C. (3632° F.), there will necessarily be a great amount of heat carried away by the water. In a reciprocating engine the metal surface with which the gas comes into contact is very small compared with that in a multiple-expansion turbine; and in a reciprocating engine the bulk of the gas may expand and fall from its maximum temperature to the temperature at exhaust without ever coming near a metal surface. In a multiple-expansion turbine, on the other hand, every particle of gas must practically slide along a metal surface immediately it comes to the first ring of blades. With turbines employing gas which enters the turbine casing at such a temperature, the heat lost through the walls and carried away by the water must necessarily be very great indeed. It is true that the metal surface in contact with the gas can be allowed to be at a much higher temperature than the inside of the cylinder walls of a reciprocating engine; but, in spite of this, the heat lost through the walls and carried away by the cooling water (or other cooling medium) will probably be much greater with a turbine actuated by gas entering the turbine casing at about 2000° C. than in a reciprocating engine in which the maximum temperature is 2000° C. This loss of heat will cause the actual work done by the engine to be very much below the ideal. This is not only important in itself, but, as will be explained subsequently (pages 50 and 51), it prevents useful employment of a high ratio of negative work to gross work. The question of utilizing this lost heat will be discussed later on pages 59 to 66.

Cycle I, Case 3a.

Instead of employing cooling arrangements for the metal, some or all of the available heat energy of the gas can be converted into kinetic energy before causing it to act on the turbine, so that the latter is not exposed to an unduly high temperature. This can be done by allowing the gas, when at the maximum temperature, to expand in a divergent nozzle till its temperature falls to a degree that the turbine can stand. More than one nozzle can be employed, but, to reduce the radiation losses, the nozzles should be large and few in number.

Suppose that the gas is compressed adiabatically to 300 pounds absolute, and then is heated at constant pressure to a temperature of 2273° absolute C. (4132° F.), as in Case 3. If now the gas be allowed to expand in a suitable nozzle, adiabatic expansion can be obtained; and if this be continued till the pressure falls to 15 pounds absolute the temperature will be 966 absolute C. (693° C.). This is just below the temperature which was fixed on as a maximum for a turbine without artificial cooling. The entropy-temperature diagram will be the same as in Case 3, Fig. 18 (page 43), and E will therefore be the same, namely 0.58. The ratio of negative work to gross work will also be the same as in Case 3, namely 0.3.

Referring to the pressure-volume diagram for Cycle I Case 3, Fig. 17 (page 42), the area $zZHK$ represents the kinetic energy of the gas leaving the nozzle, which kinetic energy equals 33,840 foot-pounds. This is for a quantity of gas which measures 1 cubic foot at A. The velocity is 5290 feet per second.

For the sake of comparison it may be advantageous to mention the velocities of the steam jets employed in De Laval steam turbines. If saturated steam at 50 pounds, absolute pressure is expanded adiabatically to a pressure of 0.6 pounds absolute, which corresponds to a temperature of 85° F., and its heat energy turned into kinetic energy, the velocity acquired works out at 3690 feet per second. If saturated steam at 300 pounds absolute pressure were treated similarly, the velocity would be 4380 feet per second. The velocities actually obtained in practice must be somewhat less than these figures, owing to friction in the nozzles.

To get the best results from a fluid velocity such as 5290 feet per second would require, with a single turbine wheel, a vane speed which cannot be obtained at present for want of a sufficiently strong and light material—the stresses produced by centrifugal force are too great. This difficulty is experienced with De Laval turbines. The obvious way out of the difficulty is to employ several wheels in series, the gas passing through the several wheels with diminishing velocity, but with nearly constant pressure. This has been done in steam turbines.

With the same object of reducing the vane speed, a device has been proposed whereby the nozzles are mounted on a wheel which rotates in the opposite direction to the wheel carrying the vanes. If the two wheels rotate at the same speed (in opposite directions) this speed will be half of that of the single wheel if the nozzles were stationary. The centrifugal force is, therefore, only one-fourth of what it would otherwise be.

The frictional losses in the nozzles of a gas turbine will probably be less than those in the nozzles of a steam turbine for the same velocity of exit from the nozzle.

Cycle I, Case 4a.

If one tries to work to the same entropy-temperature diagram as in Case 4, Fig. 20 (page 45), but employs a divergent nozzle, as in Case 3a, to reduce the maximum temperature to 700° C., so that the gas can be used in a turbine without cooling arrangements, T_1 in this case will be 725° absolute C. (452° C.). It is not necessary, therefore, to perform all the adiabatic expansion in a divergent nozzle, but a portion of it can be performed in the turbine. If the fluid is expanded in the nozzle only till its temperature falls to 700° C. (1292° F.), the pressure will then be 42 pounds absolute; so that 27 pounds can be dropped in the turbine.

Referring to the pressure-volume diagram for Cycle I, Case 4, Fig. 19 (page 44), the line qQ is drawn to represent the pressure at which the gas leaves the nozzle. The kinetic energy of the gas leaving the nozzle is represented by the area $XMQq$. It can be ascertained that this amounts to 33,660 foot-pounds (for one cubic foot of gas measured at A), and the velocity works out at 5280 feet per second. E will be the same as in Case 4, and so will the ratio of negative work to gross work.

It seems to the author that an engine working on this cycle, according to Case 3a or Case 4a, or between these, has good prospects. The ideal efficiency is high—from 0.58 to 0.68. How near one could approach this efficiency in practice would depend, of course, both on the losses in the motor proper and on the losses in the pump.

The losses in the motor proper may be taken to include the losses in the combustion chamber, if such is employed, and in the nozzles. The motor losses will then consist of:—

1. Loss of heat by radiation and conduction.
2. Fluid friction.
3. Friction in turbine bearings.
4. Loss due to incomplete expansion.

The first loss will be large, but should be less than in reciprocating engines, owing to the higher velocities employed and to the higher temperatures allowable in the metal.

The second loss will be considerable, but much less than in turbines using saturated steam. It has been found by experiment that hot dry air causes much less friction than wet steam. (The steam is always wet in a De Laval turbine casing, unless it enters the nozzles with a large amount of superheat.)

The third loss will be trifling and the fourth loss should be moderate. The discharge of heat with the exhaust gases is here only considered as a loss in so far as it exceeds that of an ideal engine.

It is difficult to estimate the pump losses. Rotary compressors on the turbine principle seem to have been employed up to only about 80 pounds pressure. Whether or no they are suitable for high pressures is a point which it is very desirable to ascertain. One would be inclined to believe that the fluid frictional losses with such machines would be very great if attempts were made to obtain high pressures. It by no means follows, however, that a fairly efficient rotary air compressor cannot be devised.

A reciprocating compressor always has the disadvantage that the air when drawn in becomes heated by contact with the hot metal surfaces before compression commences. This evil is reduced by compounding. It is an evil which occurs to a serious extent with reciprocating gas engines working on the Otto cycle.

With a reciprocating compressor it will be difficult to avoid the necessity of jacketing the cylinder if high compressions are employed. This will bring the compression curve below the adiabatic and reduce the efficiency as before explained.

In any case, whatever be the nature of the pump, there is bound to be a certain amount of heat passed through the walls of the pump cylinders or casing. If this loss be made up by friction or impact within the pump, the compression may be along an adiabatic curve, but the loss will still have to be considered.

The ratio of negative work to gross work (in the particular cases here referred to) is somewhat high—0.3 to 0.4. In the case of a turbine one need not fear the increase in the bulk of the engine due to this high ratio; for the bulk of the turbine will probably be very small for the power. Frictional and other losses become, however, of much greater importance when the ratio is high. To show this forcibly, consider an extreme case. Suppose that the ratio of negative work to gross work in an ideal engine is 0.5, or, in simpler language, suppose the pump requires half the gross power of the machine, there being no friction. If now the machine is not ideal, and if the mechanical efficiency of the pump is only $\frac{2}{3}$ and that of the motor proper only $\frac{3}{4}$, no useful work whatever will be got out of the machine—all the work will be absorbed by friction. For, if the power of the motor proper, including that spent on friction, is 100, the pump will require 50, and as its efficiency is $\frac{2}{3}$, it will take 75. This is exactly what the motor will give out after deducting friction. There will, therefore, be no power got out of the machine. When there is a high ratio of negative work to gross work, success will, therefore, be dependent largely on the efficiency of the pump. Unless the pump is at least fairly efficient, success cannot be expected. In the Diesel engine the bulk of the air is compressed to about 500 pounds per square inch, and the air which carries the oil into the cylinder is compressed from 100 pounds to 200 pounds higher.* It would be interesting to know with what efficiency the air is compressed in the Diesel engine.

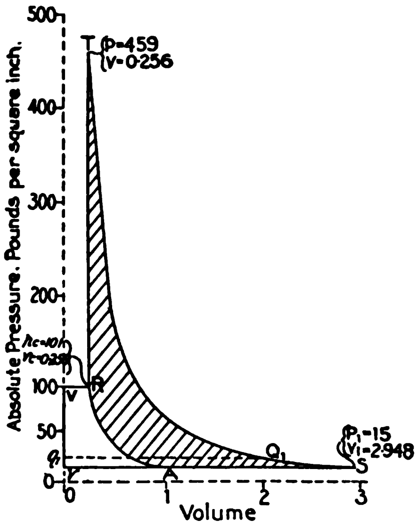


FIG. 21 —Cycle II, Case 1. Pressure-volume diagram.

Otto cycle reciprocating engines having ideal efficiencies of 0.4 to 0.45 have given practical efficiencies of half that amount. By practical efficiency is meant ratio of brake horse-power to thermal units in gas consumed, calculated on the higher calorific value. When the ideal efficiency is

* "The Diesel Engine," by H. Ade Clark. Proceedings, Inst. Mech. Engrs., 1903, Part 3, page 395.

increased above 0.45, the ratio of practical efficiency to ideal efficiency usually falls below 0.5—the greater the ideal effiiency, the greater are the losses. With a turbine the losses ought also to increase when the ideal efficiency is increased, but whether to the same extent as with an Otto engine it is

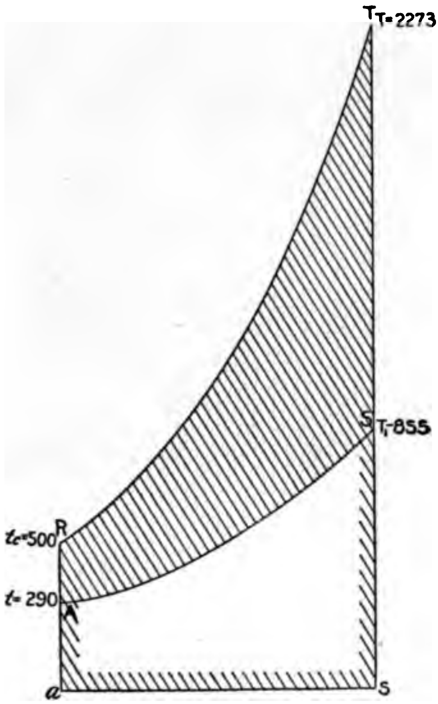


FIG. 22.—Cycle II, Case 1 Temperature-entropy diagram.

difficult to say. When considering high compressions, it is well to note that the Diesel engine, with a high compression and an incomplete expansion, has given some of the highest practical efficiencies yet attained. The compression should not cause the same trouble in starting a turbine as in starting a reciprocating engine, as with a turbine it should be practicable to arrange that at every instant the gross work is greater than the negative work. With a reciprocating engine having a single cylinder working on the Otto cycle there are, of course, periods when the negative work exceeds the gross work.

Cycle II, Case 1.

With regard to explosion turbine engines, suppose that the fluid is compressed adiabatically to, say, 101 pounds per square inch absolute, that is to a temperature of 500° absolute C. (932° F.). Let it now be heated at constant volume by explosion, and let there be a mixture of such a strength that the temperature will rise to 2000° C. (2273° absolute C.). The pressure will then be 459 pounds absolute. If the gas is now allowed to expand adiabatically till its pressure is atmospheric (when its temperature will be 855° absolute C.), and then cooled at that pressure till it resumes its original state, the pressure-volume and entropy-temperature diagrams will be as shown in Figs. 21 and 22 (pages 52 and 53).

In Fig. 22 the heat supplied to the fluid is represented by the area $aRTs$, the heat rejected by the area $aASs$, and the heat converted into work by the area $ARTS$.

Therefore $$E=\frac{\text{area } ARTS}{\text{area } aRTs}=0.55.$$

The negative work can be compared with the gross work in Fig. 21. The ratio of negative work to gross work

$$=\frac{\text{area } vVRTS}{\text{area } vVRA}=0.23.$$

Cycle II, Case 1, very nearly resembles common practice to-day with reciprocating explosion engines. The expansion is, however, continued to atmospheric pressure. This as a rule is not desirable in a reciprocating engine, on account of the extra length required to be given to the engine cylinder, which not only increases the loss by friction but increases the loss of heat by the expanding gas and, if the same length of stroke is employed for drawing in the fresh charge, increases the heating of the charge before compression. The case, however, is very different with turbines; and there seems no good reason why with these the adiabatic expansion should not be carried practically to atmospheric pressure.

In practice the maximum pressure and the average maximum temperature throughout the gas would be less than the values here indicated, owing to radiation losses.

Cycle II, Case 1a.

The gas could not be allowed into an uncooled turbine at the maximum temperature in Cycle II, Case 1; but, if the expansion was performed wholly or nearly wholly in a divergent nozzle, the temperature of exit from the nozzle would be sufficiently low to allow of the gas entering an uncooled turbine.

For example, if the gas at the maximum temperature of 2273° absolute C. (4123° F.) and the maximum pressure of 459 pounds absolute were expanded in a perfect divergent nozzle till the temperature fell to 700° C. (973° absolute C.), which was fixed on as the maximum allowable temperature in an uncooled turbine, the mean pressure on leaving the nozzle would be 23.5 pounds absolute. The kinetic energy of the gas (1 cubic foot at A) on leaving the nozzle would be represented by the area $VRTQ_1q_1$ in Fig. 21, and would amount to 20,500 foot-pounds. The mean velocity (the square root of the mean square) would be 4120 feet per second.

On comparing Cases 1 and 1a of Cycle II by reference to the Table (page 75) with Cases 2, 3, 3a, 4 and 4a of Cycle I, which have the same maximum temperature, it will be found that the efficiency is very much greater than Cycle I, Case 2; is nearly as great as Cycle I, Cases 3 and 3a; and is considerably below Cycle I, Cases 4 and 4a. The ratio of negative work to gross work is, however, greater than in Cycle I, Case 2, and less than in Cases 3, 3a, 4 and 4a of Cycle I.

There are two objections to the use for turbines of a cycle such as Cycle II, and these objections must be set against the advantage which turbines would possess over reciprocating explosion motors, in being able to make better use of the tail end of the pressure-volume diagram.

One objection is that explosions at constant volume have to take place intermittently, while a turbine desires a continuous supply of fluid. If the supply is not continuous the power of the turbine is less than it would otherwise be for a given size of machine; and the initial cost, the bulk and—most important—the loss by friction are greater in proportion to the power developed than they would otherwise be.

The other objection is that the fluid must leave the explosion chamber at varying pressure. This necessitates, unless special means are provided to prevent it, the fluid entering the turbine casing either at varying pressure or at varying velocity, which of course is objectionable, as the speed of rotation of the turbine cannot, during the period of a cycle, be made to vary correspondingly.

The second objection might be met by employing in a parallel flow turbine of the De Laval type long radial blades, and causing the nozzles to be altered in position according to the pressure, so as to direct the gas onto the outer ends of the blades at low pressures. The difficulty could also be met by an arrangement of reciprocating engine combined with a turbine, the gas being first expanded in the reciprocating engine to a certain pressure and then passed on to the turbine to complete its expansion. If several reciprocating cylinders were employed, the first objection also would be got over, but it is true that with such a combination some of the most important advantages of the turbine would be lost. The idea is, however, in the author's opinion, worthy of consideration. Reciprocating steam engines have been successfully combined with steam turbines in this manner.*

Cycle II, Case 2.

An explosion engine, in which a very high compression pressure is employed, will now be considered. If compression be carried to 818 pounds absolute as in Cycle I, Case 4, one obtains with a maximum temperature of 2000° C. (3632° F.) a maximum pressure of 2045 pounds absolute and a very high ratio of negative work to gross work. If a much lower compression—namely 417 pounds absolute—is adopted, this will give a temperature of compression of 750° absolute C. (1382° F.). Working on the same cycle as in the last case and arranging the explosive mixture to give a maximum temperature of 2000° C. (2273° absolute C.), a maximum pressure of 1265 pounds absolute is obtained, and the pressure-volume and the entropy-temperature diagrams will be as shown in Figs. 23 and 24 (page 58).

Referring to Fig. 24,

$$E=\frac{\text{area } AUWW_1}{\text{area } aUWw}=0.68.$$

Referring to Fig. 23,

$$\frac{\text{negative work}}{\text{gross work}}=\frac{\text{area } u_1U_1UA}{\text{area } u_1U_1UWW_1}=0.38.$$

E is the same as in Cycle I, Case 4, and the ratio of negative work to gross work is also the same. The compression is lower than in Cycle I, Case 4, but the maximum pressure is very much higher.

The excessively high maximum pressure is an objection to this case.

* See Paper by Professor Rateau read before the North of England Institute of Mining and Mechanical Engineers at Newcastle-on-Tyne, Dec. 13, 1902; or Paper by the same author read at the Chicago Meeting of the Institution of Mechanical Engineers, Proceedings 1904, Part 3 (page 737).

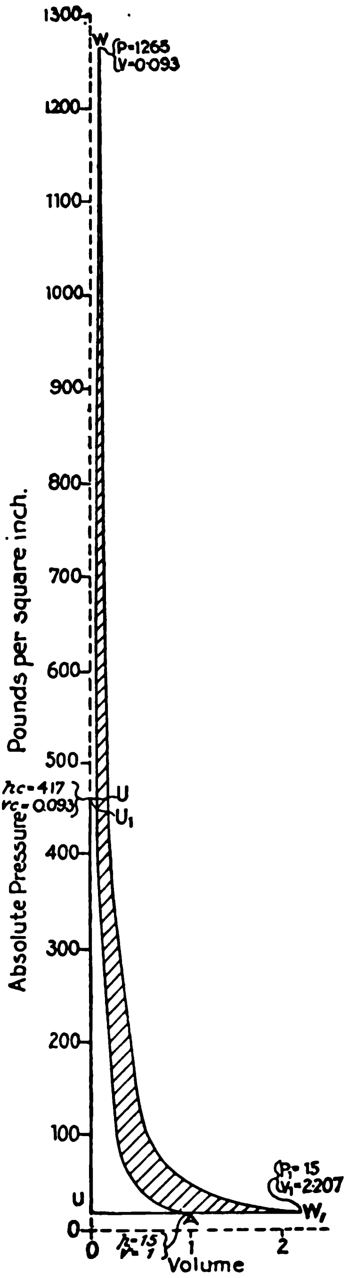


Fig. 23.—Cycle II Case 2. Pressure-volume diagram.

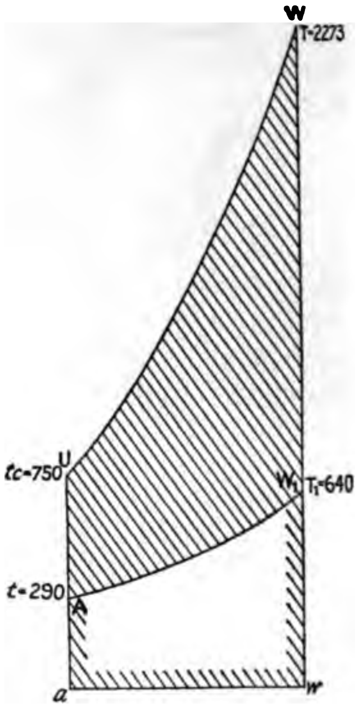


Fig 24.—Cycle II, Case 2. Temperature entropy diagram.

Cycle II, Case 2a.

If the expansion took place in an ideal divergent nozzle as before till the temperature fell to 700° C. (973° absolute C.), the gas would still have a pressure of 70 pounds absolute, while the mean velocity of exit from the nozzle would be 4300 feet per second. If the gas were expanded in the nozzle down to 25 pounds absolute, the temperature would then be 741° absolute C. (1366° F.), and the mean velocity of the gas leaving the nozzle would be 4830 feet per second.

Cycle III, Case 1.

It has been proposed, when a water-jacket is employed, to utilize the heat passed into the jacket water by causing this heat to generate steam from the water. This steam could then receive further heat from the products of combustion, which would therefore be reduced in temperature, while the steam would be superheated. The steam and products of combustion could then expand adiabatically, doing work in the same or in separate turbines. The carrying out of this idea would affect the efficiency in the several cases considered of Cycle I. Cooling arrangements are not required in Cycle I, Case 1, so this case need not be further considered. In Cycle I, Case 2, let it be supposed that the combustion chamber is jacketed and that the jacket water is heated and converted into steam by heat taken from the products of combustion, which have their temperature thus lowered from 2000° C. to 700° C., that is, to the temperature at which they can safely be allowed into an uncooled turbine, the steam being superheated up to 700° C. Let this be called Cycle III, Case 1.

Referring to Fig. 16 (page 41), the heat in the products of combustion which is converted into work is now represented by the area *ABCD* instead of by the area *ABEF*. The heat represented by the area *dCEf* has, however, been employed in heating water and generating and superheating steam. The fraction of this heat which is converted into work will not now be as great as in the original scheme of working. That is to say, the net work got out of the heat put into the water and steam will be less than the area *DCEF*. By transferring heat to the water and steam from the gas, *E* is therefore reduced. There must, however, in any case, as already mentioned (page 30), be lost in practice a large amount of heat when the products of combustion enter the turbine casing at a temperature such as 2000° C., and, by adopting this combined steam and gas scheme, a much higher practical efficiency may possibly be attained than would otherwise be possible. As the net work ideally is less than in Cycle I, Case 2, and as the negative work is not less (and may be greater by the amount of work required to pump the water into the jacket if under pressure), the ratio of negative work to gross work is increased. In Case 2 of Cycle I, the ratio of negative work to gross work is low, and it will, therefore, be allowable to increase this ratio.

Cycle III, Case 2.

Case 3 of Cycle I could be modified in the same way by reducing the temperature of the products of combustion from 2000° C. to 700° C., and by employing the heat so given up in heating water and generating and superheating steam. The steam could be generated at 300 pounds pressure absolute (the same pressure as the products of combustion) and superheated to 700° C. at this pressure. The steam and gas could then be expanded adiabatically in the same or separate turbines. As in the previous case, *E* would be reduced, and the ratio of negative work to gross work increased. As in the previous case, the practical efficiency might also be largely increased.

The pressure-volume and entropy-temperature diagrams for the gas in this case (called Cycle III, Case 2) are shown in Figs. 25 and 26 respectively (pages 61 and 62). The gas is compressed along the line AG as in Cycle I, Case 3, till its pressure is 300 pounds absolute and its temperature is 409.5° C. (682.5° absolute C.). It is then heated by combustion at constant pressure along the line GH as in Cycle I Case 3, till its temperature is 2000° C. (2273° absolute C.). Heat is now withdrawn from the gas at constant pressure and transformed to the water and steam, the temperature of the gas falling along the line HH_1, to 700° C. (973° absolute C.) at H_1. The heat transferred from the gas to the water is

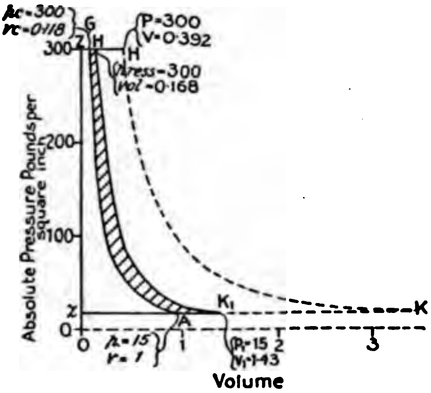


FIG. 25.—Cycle III, Case 2. Pressure-volume diagram—gas.

represented, Fig. 26, by the area k_1H_1Hk. The gas now expands adiabatically along the line H_1K_1 till the pressure is 15 pounds absolute, when the temperature will be 140° C. (413° absolute C.). The contraction of the gas at constant pressure along the line K_1A completes the cycle. Dotted lines have been placed on Figs. 25 and 26 to illustrate Cycle I Case 3, where this differs from the present cycle. The two cycles can thus be compared.

Pressure-volume and entropy-temperature diagrams for the water are shown in Figs. 27 and 28 (pages 63–65). Referring to Fig. 28, the water is heated at a constant pressure of 300 pounds per square inch absolute along the line fc from

100.6° C. (373.6° absolute C.) to 214° C. (487° absolute C.), which is the boiling point at this pressure. The water is now converted into steam, this process being represented by the line *cg*; and the steam is superheated at constant pressure as represented by the line *gd*, till its temperature is 700° C.

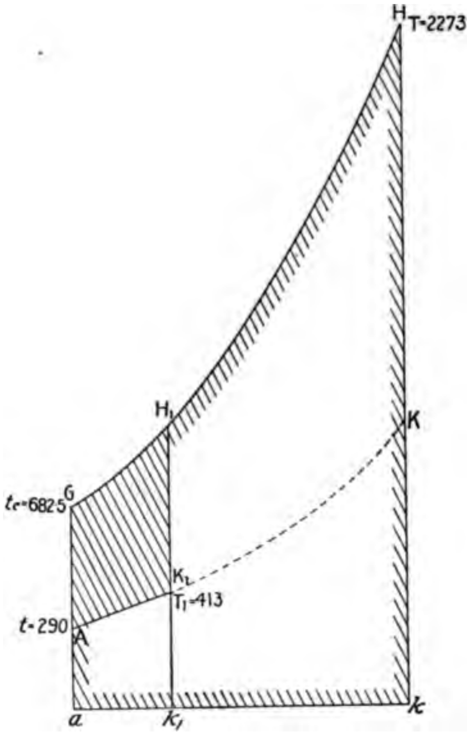


FIG. 26.—Cycle III, Case 2. Entropy-temperature diagram—gas.

(973° absolute C.). The steam is then expanded adiabatically along the line *de* till it falls to 15 pounds absolute pressure, its temperature then being 184° C. (457° absolute C.). The steam is now exhausted and cools along the line *eh*. At *h* it is saturated, its temperature being 100.6° C. (373.6° absolute C.), and thereafter it condenses along the line *hf* and is compressed to its initial state.

Fig. 27 shows the work done by the steam in its generation, superheating and adiabatic expansion. The work done in forcing the water into the chamber at 300 pounds pressure is not shown in Fig. 27 and is negligible in the present investigation.

The heat required to raise the water from 373.6° absolute C. to 487° absolute C., is represented in Fig. 28 by the area f_1fcc_1. The area c_1cgg_1 represents the latent heat of steam at a pressure of 300 pounds absolute (the temperature being 487° absolute C.), and the area g_1gde_1 represents the heat re-

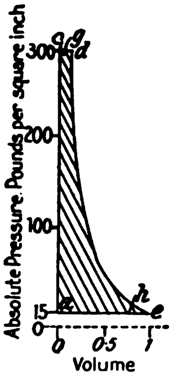


FIG. 27.—Cycle III, Case 2. Pressure-volume diagram—steam.

quired to superheat the steam from 487° absolute C. to 973° absolute C. The area f_1fhh_1 represents the latent heat of steam at a pressure of 15 pounds absolute, and the area h_1hee_1 represents the heat required to superheat this steam from 373.6° absolute C. to 457 absolute C.

Comparing this case with Case 3, of Cycle I, it is found that the total heat absorbed is the same in both cases, being represented by the area *aGHk* in Fig. 26. The portion of this heat which is converted into work in Case 3, Cycle I, is represented by the area *AGHK*, while the corresponding portion in the present case is represented by the sum of the areas AGH_1K_1, Fig. 26, and *fcgdeh*, Fig. 28. This sum is less than the area $AGHK$, and E in this case is only 0.33 as compared with 0.58 in Case 3 of Cycle I. The fall in the value of E is due to the relatively low efficiency of the steam portion which has an ideal efficiency of only 0.28. (For a steam engine this is really not low.)

The feed-water has been taken at a temperature corresponding to atmospheric boiling-point. It has been assumed that the steam is exhausted into the atmosphere, and is not condensed for use over again. It would, therefore, be necessary, in order to follow the cycle, to heat the feed-water to 100° C. It should not be difficult to approximately accomplish this by utilizing the heat of the exhausting gases. By heating the feed-water still more, the efficiency could be improved; but the improvement would be slight (less than in an ordinary steam engine) and the feed-water would have to be under pressure. As, moreover, any increase of exhaust or back pressure is a serious matter with a turbine, and as feed-water heaters must to a certain extent affect this back pressure, any prospect of gain by heating the feed-water beyond 100° C. need not be considered.

The gross work in the present case is represented, Figs. 25 and 27, by the area zZH_1K_1 + the area *acde*. This is less than the gross work in Cycle I, Case 3, which is represented by the area $zZHK$. The negative work in Cycle I, Case 3, was represented by the area $zZGA$. In the present case it is also represented by this area, neglecting the work of pumping the water into the jacket. The ratio of negative work to gross work in the present case is 0.41 as compared with 0.3 in Cycle I, Case 3. This ratio (0.41) is rather high. It will, however, probably not be so objectionable in the present case as the ratio 0.40 in Case 4 of Cycle III, as the real efficiency in practice will come nearer to the ideal in this case than in Case 4 of Cycle III. In the present case the ratio could be reduced by lowering the compression. This would reduce E.

As the mass of the water employed is not the same as the mass of the air and fuel, the scale for entropy in Fig. 28 has been made different from that in the other entropy-temperature diagrams, so that in all these diagrams areas represent quantities of heat to the same scale. In all the pressure-volume diagrams the scales are the same except in Fig. 29 (page 66), which will be referred to hereafter. It might be mentioned here that all the numerical results given in this Paper have been obtained by calculation and not by scaling the diagrams.

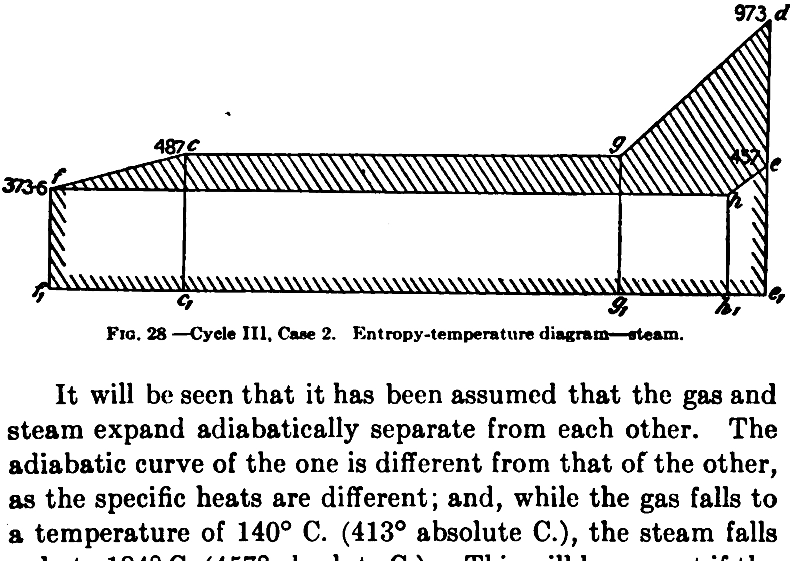


Fig. 28 —Cycle III, Case 2. Entropy-temperature diagram—steam.

It will be seen that it has been assumed that the gas and steam expand adiabatically separate from each other. The adiabatic curve of the one is different from that of the other, as the specific heats are different; and, while the gas falls to a temperature of 140° C. (413° absolute C.), the steam falls only to 184° C. (457° absolute C.). This will be correct if the steam and gas are not mixed. It is much simpler to consider this case than to consider the case where the gases are intimately mixed. In this latter case the diagram Fig. 26 would be altered, and it could not so easily be seen where the loss of efficiency came in. In practice, however, it will probably be found convenient to mix the gases. This will alter the diagrams and the efficiency somewhat; but what has been considered gives a good idea of the general effect of the employment of steam in conjunction with gas. If the steam and gas are not mixed, a condenser could be employed for the former. The steam could then be expanded to a much lower temperature and pressure, and the efficiency would be considerably raised.

Cycle II could be modified by combining steam with the gas, in the same way as Cycle I was modified. A case of this nature has not been worked out; but Case 1 of Cycle II could probably be modified in this way. Case 2 of Cycle II could not be so treated on account of the high ratio of negative work to gross work that would occur.

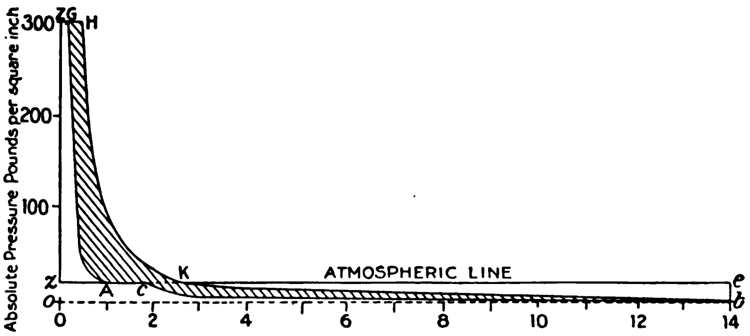


Fig. 29.—Pressure-volume diagram. The horizontal scale is half that of the other diagrams.

One might try to improve on all these cycles, by extending the adiabatic expansion line of the gas below atmosphere, instead of stopping it at atmospheric pressure. It would, of course, be necessary to compress the fluid back again to atmospheric pressure; but, if this compression were isothermal or between the isothermal and adiabatic, there would be an increase of efficiency. Carnot's cycle is in fact being approached in the lower part of the diagram.

Figs. 29 and 30 are respectively pressure-volume and entropy-temperature diagrams of Cycle I, Case 3, modified by continuing the adiabatic expansion to a pressure of 2 pounds per square inch absolute. The scale for volumes in

Fig. 29 has, for convenience, been made half that of the other diagrams. *Kb* represents the addition to the adiabatic line of expansion, and *bc* represents isothermal compression of the gas from 2 pounds absolute at *b* to 15 pounds absolute at *c*.

There should be no difficulty in a turbine in extending the expansion from *K* to *b*. There may be difficulty, however, in

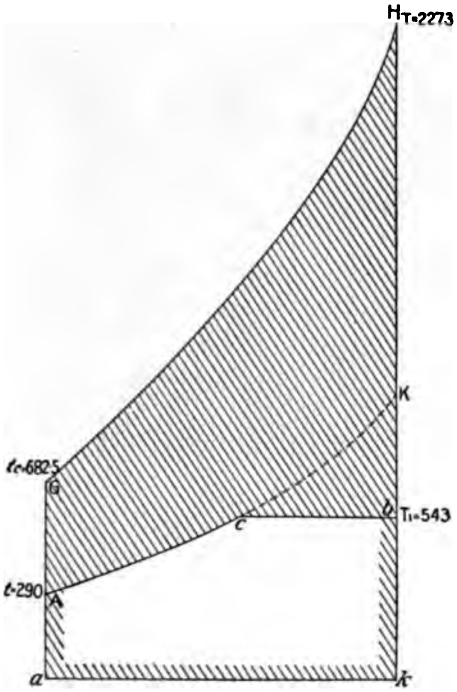


FIG. 30.—Entropy-temperature diagram.

getting isothermal compression from *b* to atmospheric pressure at *c*. As the volume at *b* is 14 times the initial volume it will be desirable to get the fluid discharged as quickly as possible. A rotary compressor will probably be best for this purpose. A compression, sufficiently near to the isothermal and sufficiently remote from the adiabatic to raise the efficiency appreciably, should be obtainable.

The temperature at b is 270° C. (543° absolute C.), and if the compression were isothermal, this would of course be the temperature all along the line bc. The gases could be passed through or around water-cooled tubes to keep down the temperature during compression. With the gas at a temperature of 543° absolute C. it would not do to spray water into it, unless sufficient water were sprayed to cool the gas below the boiling-point of the water, which is 326° absolute C. at this pressure.

If compression takes place along the isothermal line bc, a net amount of work will be gained, represented by the area Kbc. The gas will be discharged into the atmosphere at c, the volume at discharge being 1.874 of the original volume (at A). Even if the compression is not isothermal, an amount of work may be gained which will wipe out the extra losses in the machine, provide for pumping out the cooling water, and perhaps leave a margin of net gain.

In Fig. 30 the heat absorbed by the fluid is represented by the area $aGHk$, the heat rejected by the area $aAcbk$, and the heat converted into work by the area $AGHbc$. As the heat absorbed remains unchanged, while the heat converted into work is increased by the area Kbc, E is of course increased.

$$E = \frac{\text{area } AGHbc}{\text{area } AGHk}.$$

This enlarging of the diagram of course affects the ratio of negative work to gross work. Referring to Fig. 29 (page 66),

$$\begin{aligned} \text{gross work} &= \text{area } zZHKebc \\ \text{negative work} &= \text{area } zZGA + \text{area } Keb \\ \text{net work} &= \text{area } AGHbc \end{aligned}$$

$$\frac{\text{negative work}}{\text{gross work}} = \frac{\text{area } zZGA + \text{area } Keb}{\text{area } zZHKebc}.$$

In the free piston explosion engines, which were at one time in fairly common use, the best known of which is the

Otto and Langen, the expansion was carried to a pressure considerably below the atmosphere. The compression to atmospheric pressure which followed must have been between the isothermal and the adiabatic.

If this continuation of the adiabatic expansion below atmospheric pressure is not found to be advisable to the extent that has just been described, it may be found advisable to a less extent. If it is found advisable in any case, it is more likely to be so in a case in which the high pressure of the gases after combustion is reduced to a low pressure in divergent nozzles, before the gas is allowed into the turbine casing, than in a case in which the whole fall of pressure takes place in the turbine casing. In the former case very high vane speeds are necessary, and the friction between the rotating parts and the fluid in the casing is an extremely important matter. The reduction of the pressure within the turbine casing from atmospheric pressure (or above that) to one-quarter or one-eighth of that amount may therefore very much reduce the frictional losses. It is true that the rotary pump, if such is employed for completing the cycle, has to deliver at atmospheric pressure, but the rotating parts of the pump can revolve at a much lower speed, and the friction will therefore be of much less consequence.

With such high speed turbines there is another question to be considered. It has been stated in discussing Cases 3a and 4a of Cycle I, and 1a and 2a of Cycle II, that the velocity of the gases escaping from the divergent nozzles would be over 4000 feet per second, if the heat energy converted into kinetic energy was as mentioned. The author is not, however, aware of any results of experiments having been published in which velocities of these amounts were obtained, when the pressure of the medium into which the divergent nozzle discharged was atmospheric. It is supposed by some that there is a maximum limit to the velocity of a gas leaving a divergent nozzle and escaping into a given medium which is at a given pressure, etc., and that this limit velocity is dependent on the pressure in the medium into which the nozzle discharges, and is less when the pressure in this medium is greater, and *vice versa*. That is to say, it is supposed by some that, after a certain velocity of discharge has been attained, no increase in the initial temperature or pressure will increase this velocity; but a reduction of the pressure in the medium may do so. The author does not express any opinion himself on this point, but if it should be found that the reduction of the pressure inside a turbine casing below atmospheric pressure enables the heat energy of the gas to be more effectively converted into kinetic energy, this will be a further argument in favor of so reducing the pressure. Whether or not there is an advantage to be gained remains to be proved, but there is at any rate a possibility of gain by thus extending the expansion and it is a possibility which, in the author's opinion, should not be ignored. In dealing with large volumes and small pressures there is, as already mentioned, an immense difference between turbines and reciprocating engines.

Cycle IV.

The fourth cycle which will be considered in this Paper is one in which a high ideal efficiency can be obtained with a low compression, and without having an abnormally high ratio of negative work to gross work.

Figs. 31 and 32 are, respectively, pressure-volume and entropy-temperature diagrams for an engine working on this cycle. In explaining the cycle it is best to start at E^1. At this point the temperature of the fluid is 1592° C. (1865° absolute C.), and the pressure is 30 pounds absolute.

Let the fluid be heated by combustion at constant pressure along the line E^1C^1 till the temperature reaches 2000° C. (2273° absolute C.). Now let the gas expand adiabatically from C^1 to D^1 till the pressure is atmospheric. The temperature will then be 1592° C. (1865° absolute C.). Now let the gas pass through a regenerating chamber and be cooled at a constant pressure from D^1 to F^1 till the temperature is 80° C. (353° absolute C.). The gas escapes at F^1 into atmosphere, and thereafter cools at constant pressure to 17° C. (290° absolute C.) at A. A new charge is taken at A and compressed adiabatically to B^1 where the pressure is 30 pounds absolute and the temperature 80° C. (353° absolute C.). The fluid is now passed through the regenerating chamber, and is heated at constant pressure along the line B^1E^1, taking back the heat given up by the last charge. This will raise its temperature to 1592° C. (1865° absolute C.) and place the fluid in the condition it was at the start.

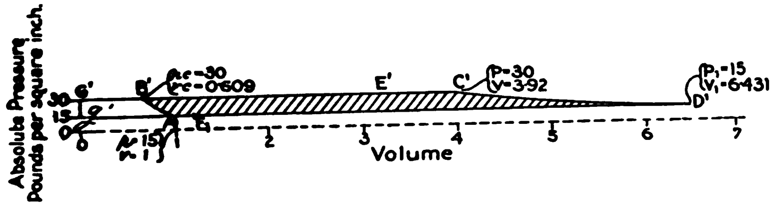


FIG. 31.—Cycle IV. Pressure-volume diagram.

Referring to Fig. 31, the gross work is represented by the area $g^1G^1C^1D^1$, the negative work by the area $g^1G^1B^1A$, and the net work by the area $AB^1C^1D^1$.

$$\text{Therefore } \frac{\text{negative work}}{\text{gross work}} = \frac{\text{area } g^1G^1B^1A}{\text{area } g^1G^1C^1D^1} = 0.16 \ (0.1553).$$

The heat absorbed by the fluid (other than that obtained in the regenerator from a previous charge) is represented in Fig. 32 by the area eE^1C^1d. The heat rejected (other than that given to the regenerator) is represented by the area aAF^1f. The heat converted into work is represented by the difference of these two areas.

$$\begin{aligned} \text{Now area } AB^1C^1D^1 &= \text{area } aB^1C^1d - \text{area } fF^1D^1d - \text{area } aAF^1f \\ &= \text{area } aB^1C^1d - \text{area } aB^1E^1e - \text{area } aAF^1f \\ &= \text{area } eE^1C^1d - \text{area } aAF^1f. \end{aligned}$$

Therefore the area $AB^1C^1D^1$ represents the heat converted into work—

Therefore $$E=\frac{\text{area } AB^1C^1D^1}{\text{area } eE^1C^1d}=0.84.$$

The ideal efficiency is high; but the highest actual efficiency which could practically be obtained would be very

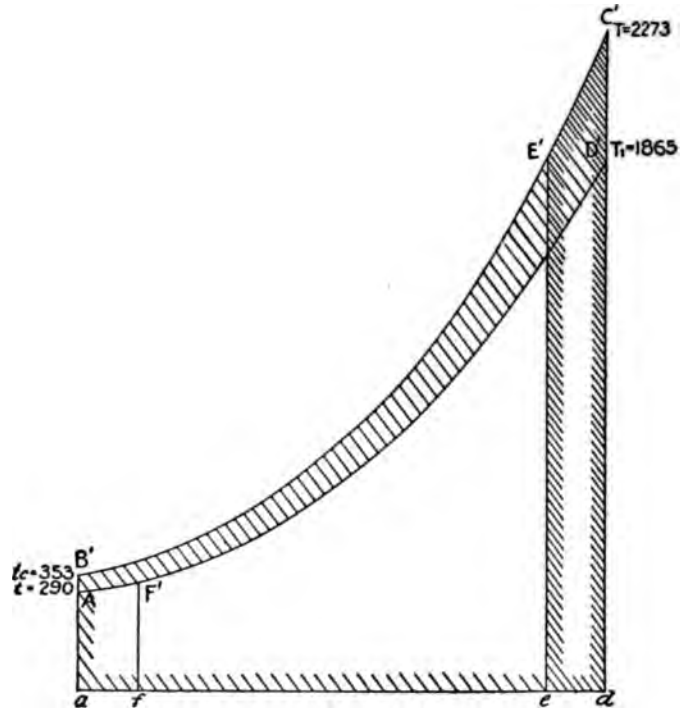


FIG. 32.—Cycle IV. Entropy-temperature diagram.

much below this. Besides the losses in the motor proper and in the pump, there would be a very great loss in the regenerator. It would not be practicable to reduce the temperature of the exhausting gases in the regenerator to 80° C., or to raise the temperature of the fresh gases in the regenerator to 1592° C.

If the losses in the regenerator and in the passages leading to it and from it amounted to 50 per cent. of the heat which is ideally given to or taken from the regenerator, these losses would have to be made up by extra heat given to the fluid by combustion, and the efficiency would fall to 0.3. This does not take into account the losses in the motor proper and in the pump. The heat losses in the motor would probably be very great. This cycle may, however, when used with turbines, give results sufficiently good to justify its use. It certainly seems to promise better results with turbines than with reciprocating engines, on account of the lower frictional losses that might be expected with turbines. With reciprocating engines the large volumes and the low pressure of the fluid would cause extremely high percentage losses in friction.

The cycle has the disadvantage that gas at a very high temperature has to be conveyed from the regenerator to the turbine. This practically makes it absolutely necessary to have the turbine quite close to the regenerator. It would seem to be expedient to build the regenerator of brick-work and to erect the turbine in this brick-work. This would very much limit the usefulness of the cycle, as it would not be quite feasible in many cases to have either a regenerator of the nature required at the place where power is wanted, or to transmit the power from a place suitable for holding the regenerator. Nevertheless there will be cases in which it will be quite practicable to build a regenerator beside the turbine, and this cycle therefore seems to be worthy of consideration. A rotary pump driven from the turbine spindle could easily be used for compressing the gases, thus simplifying the mechanical moving parts.

The several cases can be compared in the Table (page 75). It will be seen that a high ideal efficiency is, as a rule, accompanied by a high ratio of negative work to gross work. Cycle 4 is, however, an exception to the rule. The cycle has the highest ideal efficiency and the lowest ratio of negative work to gross work. As has been already pointed out, however, the efficiency which could be actually looked for with this cycle would be very much below the ideal, and the cycle has other objections, as already stated.

Cycle I, Case 1, has a high ratio of negative work to gross work, although the efficiency is the lowest. This is because all the heat is supplied to the gas at a comparatively low temperature.

Engineers interested in any particular cycle can work out other cases for themselves if they consider it necessary, but it is suggested that after a careful perusal of this Paper the effect of any change can be guessed at with fair accuracy. It might be possible to use the exhaust gases from a turbine working according to Cycle I, Cases 1 and 3, or Cycle II, Case 1, to heat the fluid after compression and so to save fuel. Consider Fig. 14 (page 38). The heat supplied to the fluid is represented by the area *aBCd*. Part of this heat might be obtained from the hot exhaust gases and the efficiency of the cycle thus raised, as will be clear from the description just given of Cycle IV. With Cycle I, Cases 1 and 3, and Cycle II, Case 1, there should be no necessity to use a regenerator of brick or such like refractory material. The exhaust gases could be passed through tubes and the fresh air passed over the outside surfaces of the tubes, or some equivalent construction might be employed.

Many other cycles or modifications of cycles might have been investigated; but the author had considered it inadvisable to burden the Paper with them.

As regards the Carnot cycle, an engine working on this cycle would have the same value for E as one working on Cycle I for the same values of t and t_c; and, if the isothermal expansion were carried far enough, it would (under ideal conditions) do the same work per cycle for the same amount of fluid, and have the same ratio of negative work to gross work. The maximum volume of the fluid would, however, be very much greater, and although this is not such a serious matter with a turbine as with a reciprocating engine it is nevertheless not a condition to be accepted without due recompense.

TABLE—*Comparing the Several Cycles and Cases.*

Cycle.	Case.	Compression.		Maximum Temp. absolute C.°	Maximum pressure lbs. per sq. in. absolute.	Ideal efficiency. (E)	Ratio of negative work to gross work.
		Temp. absolute C.°	Pressure lbs. per sq. in. absolute.				
I	1	389	42	973	42	0.25	0.40
I	2	389	42	2273	42	0.25	0.17
I	3 & 3a	682.5	300	2273	300	0.58	0.30
I	4 & 4a	909	818	2273	818	0.68	0.40
II	1 & 1a	500	101	2273	459	0.55	0.23
II	2 & 2a	750	417	2273	1265	0.68	0.38
III	2	682.5	300	2273	300	0.33	0.41
IV	—	353	30	2273	30	0.84	0.16

If an engine working on Cycle I has $T_1 = t_c$, then an engine working on the Carnot cycle with the same values of t and t_c would require, in order to do the same work, to commence its adiabatic expansion at the point where the engine working on Cycle I leaves off. If p and P_1 on the Cycle I engine are atmospheric pressure, then, on the Carnot cycle engine, the whole of the adiabatic expansion would take place below atmospheric pressure. It is interesting to compare the Carnot cycle with other cycles, but it hardly seems useful in the present investigation to devote any more space to this comparison.

Although the practical efficiency of an engine is usually of great importance, there are many occasions on which a very poor efficiency will be tolerated if other conditions are satisfactory. Small gas engines using lighting gas, costing 2*s*. to 3*s*. per 1000 cubic feet, are employed in great numbers, although the fuel cost per brake horse-power hour is very high.

Small electro-motors are extensively used consuming current which costs over 2*d.* per Board-of-Trade Unit (kilowatt-hour). The fuel cost and the energy cost, respectively, in the two cases are high; but the user prefers to put up with this, rather than employ a power plant which has a higher initial cost or which requires more attention or is generally more inconvenient.

If, therefore, small gas turbines could be sold at a low price, and if they required little attention and did not readily get out of order, they might be in great demand, even although the gas consumption per brake horse-power hour was high. With the average user of a small engine, producing say 100 brake horse-power hours per week, a reduction of 10£ in the initial cost is of more consequence than a reduction of 2 cubic feet per brake horse-power hour in gas consumption. The same user would no doubt be quite willing to allow an additional 5 cubic feet of gas per brake horse-power hour if he were saved trouble and anxiety and small expenses in the working of the engine.

To produce a gas turbine cheaply it seems necessary to avoid reciprocating parts entirely and to be content with a low compression. Cycle I, Case 1, appears to lend itself to cheapness of construction and simplicity, but it might be advisable to reduce the compression at the expense of efficiency. A rotary pump could undertake the compression.

In many cases the vibration of a reciprocating engine is extremely objectionable; and a motor that ran with practically no vibration would be popular, even if its initial cost were greater and it were more extravagant with fuel. In motor cars, for example, oil or spirit explosion engines are used for their lightness and compactness; but the vibration they cause is objectionable. If a satisfactory turbine were obtainable, there is no doubt that motor-car builders would eagerly buy it and install it on their cars, even if the cost were greater and the efficiency less than the many arrangements of explosion reciprocating engines now in use.

One might mention many other uses to which gas turbines could advantageously be put, if they were obtainable as fairly efficient and reliable machines. In many factories and engineering works, electric motors fed by current from a central station are used to drive individual machines or groups of machines, in order to save the losses and inconveniences produced by driving by belts or ropes. Gas engines (reciprocating) have been used to a limited extent for the same purpose; but the foundations required for these and the vibration caused by them have prevented their extensive use. If a gas turbine were obtainable which could be set down anywhere like an electric motor, it would serve splendidly for this purpose, and, in order to displace electric driving, it would only require to possess an efficiency greater than the efficiency of the central station engine multiplied by the efficiencies of dynamo, mains, and motor.

Suction gas producers are coming into extensive use, and by their means gas engines can take the place of steam engines where they otherwise would not. Whatever objections there might be to the supply of gas to gas engines from these suction producers, these objections should be less rather than greater with turbines than with reciprocating engines.

The power of a gas turbine could be effectively varied with an insignificant variation of speed, by cutting out one or more of the nozzles or admission ports which admit the fluid to the turbine buckets or first set of buckets when several are employed in series, the fluid being passed through the acting ports or nozzles at a uniform pressure and with a uniform velocity. To enable the acting nozzles or admission ports, no matter how many may be in use, to deliver the fluid always in the same uniform manner, it will be necessary, if one flame supplies all the nozzles or admission ports, to control the fuel and air supplies to the flame in cojunction with the mechanism which cuts off the nozzles or admission ports. It will, however, be usually advisable to supply the air and fuel to the flame at a constant velocity which, although it could be done when one flame supplied a varying number of nozzles, would involve complications; and it may therefore be found expedient to have a separate flame and a separate combustion chamber for each nozzle. This would involve the necessity of igniting or extinguishing a flame for each change of power, and, although this could be done automatically, it is objectionable. Much careful consideration will be required to determine which is the least evil to put up with and which course had best be taken.

The author hopes that this comparison of the several cycles treated will be of some use in showing what may be expected from each, and which will be best suited for a motor which is required to work under given conditions and which it is important should have given characteristics. One cannot, however, compare the several cycles and estimate the actual efficiencies, &c., which might be expected in practice in anything like so satisfactory a manner as could be desired, without having more information obtained by experiment on the following three points:—

1. Losses in pneumatic compression to high pressures.
    (a) With reciprocating compressors.
    (b) With rotary compressors.
    (c) With a combination of reciprocating and rotary compressors.
2. Expansion of hot gases in divergent nozzles.
3. Radiation losses and transference of heat from gases to metals at high temperatures.

It would immensely aid the solving of the gas turbine problem if a thorough set of experiments on these three points were made and the results published. This would naturally cost a considerable amount of money; but the information obtained by the engineering world would be very good value at the price. Money has been spent and is being spent by engineering and scientific societies on investigations which, while no doubt interesting and instructive, are not of so far-reaching importance as experiments which would materially aid in the production of a successful gas turbine.

Discussion.

The President said that the author of the Paper had been for a long time associated with Mr. Dugald Clerk. It was rather an exception for the Council of the Institution to accept a Paper based upon theoretical considerations alone, the rule being to reject Papers unless they were founded upon something that had actually been accomplished in a concrete form. But he thought the large audience before him that evening justified the Council in having admitted the present Paper for theoretical discussion. With the American Society of Mechanical Engineers no formal vote of thanks was passed for a Paper. Thus time was saved for the technical discussion. The procedure of this Institution had latterly followed the same course, and the appreciation of the present Paper might now be indicated by a round of applause.

Mr. Neilson said there were many methods of working and devices suggested in the Paper which had been previously proposed, some of them over and over again. It was extremely difficult in many cases to ascertain to whom the credit was really due for those devices or methods of working, and therefore he had in most cases refrained from giving acknowledgment to any one. He mentioned that fact in case any one should feel aggrieved at not being given the credit of something stated in the Paper. He wished to emphasize what he had said at the beginning of the Paper—that its most important object was to draw opinions from other engineers who had studied the question, and especially from those who had conducted experiments.

Mr. Henry Davey, Member of Council, said he had made some experiments, but they had been on a small scale, and, with a view to starting the discussion, he would just state what his experience had been. First of all he would like to tender his thanks to the author for bringing the Paper before the Institution. It dealt with an important subject. When a Paper was brought before engineers, and the theory of the subject was put before them so lucidly as it had been put by the author, it encouraged them to look more minutely into the matter, and to see if some practical good might not come out of it. The Paper was valuable as indicating the

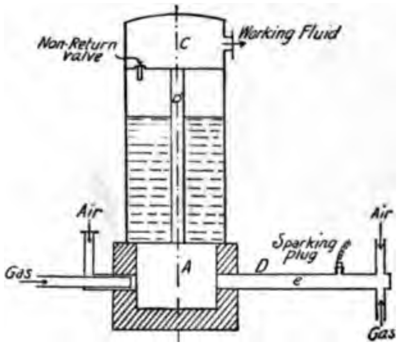


Fig. 33.—Experimental working fluid producer.

direction in which the possibilities of the problem lay. The first part of the problem was that of producing in a practical way the working fluid; then the temperature difficulties followed, and they must be succeeded by the mode of application to the turbine.

It was from the point of view of producing the working fluid that he had made two or three experiments, on quite a small scale. They were undertaken with an apparatus, shown in Fig. 33, based on Cycle III, Case 2 (page 60). *A* was a combustion chamber lined with fire-brick, and above it was a small steel boiler consisting of a shell containing the water, having fire-tubes *B* extending to the smoke-box or vessel C. A non-return valve opened into the smoke-box. When steam was raised above the pressure in the furnace, it passed through the valve and mixed with the products of combustion. The object of the apparatus was to produce a mixture of steam and products of combustion, the steam being highly superheated. The furnace was fed by means of gas and air pumps, of the relative capacities for delivering a burning mixture. The gas and air were delivered in separate pipes and came together just inside the furnace. To start the apparatus it was necessary to get the fire-brick lining into a red-hot state, so that it might maintain combustion; then the air and gas pumps were put to work. He experimented with this apparatus on a small scale, but he found many practical difficulties. He commenced with low pressures, intending to go on step by step to higher and higher pressures; but what happened was that, if the gases ceased to burn in the furnaces from any accidental cause during one or two strokes of the pump, then a big explosion frequently ensued. It seemed to him that the system promised well, if the practical difficulties of keeping the combustion constant could be overcome. If the divergent nozzle was as efficient as one might imagine, it might be that a turbine worked with a fluid produced from an apparatus of that kind, at moderately low pressure, would give a good efficiency; but the loss of efficiency in the air and gas pumps would be considerable. His pumps were reciprocating; and the loss would be probably much more considerable if they were rotary ones. That would tell very much against the scheme, as it was all a question of what useful work could be got out of a given quantity of gas. Instead of bringing air and gas from the furnace into the combustion chamber to be burnt with a constant flame, he altered the apparatus as shown at D. The gas and air united together in the pipe e and pushed forward the products of combustion of the last charge, forming a kind of cartridge in the pipe. Then the supply was cut off, and a sparking plug at once fired the cartridge. With a moderate sized brick-lined chamber A, a fairly constant pressure could be kept up. The cartridge might be small and fired very frequently, or three or four of them might be used and fired intermittently, to avoid intermittent impulses on the turbine. He had not got beyond what might be termed laboratory experiments, but had succeeded in getting the thing to work fairly at a low pressure. He had not used the working fluids to propel a turbine, but he had taken them, in the particular case he mentioned, into a reciprocating engine. It was desirable to know what were the losses due to compression both with reciprocating and rotary pumps, and then came the very great difficulties of using high temperatures in the turbine itself. With regard to the divergent nozzle, there was, as far as he knew, scarcely any practical information available. It was one of those subjects which required to be thrashed out experimentally, and one of the most important points in connection with the development of the gas turbine.

Professor F. W. Burstall said it was a source of great gratification to him to be able to attend and—he would not say discuss the Paper, because it was not a Paper which admitted of a great deal of discussion—speak on a subject which was one of immense possibilities and potentialities. It might not be for the present generation to inherit the gas turbine, but it was probable that it would come to subsequent generations. If he might say one word of criticism in regard to the Paper, he thought perhaps the author was a little optimistic on the subject. He knew very well the enormous difficulties which stood in the way of a commercial gas turbine. From the days of Watt it had taken nearly 120 years to develop a comparatively simple thing like the steam turbine, and, when it was considered that the gas turbine had not got to turn the energy of the fluid into kinetic energy on the shaft, but had to compress the fluid, to ignite the fluid, and deal with a fluid which was infinitely more troublesome to deal with than steam, it would be seen what difficulties had to be overcome. He did not for a moment say that a gas turbine could not be made; perhaps he would plead guilty to having schemed a good many himself. There was no difficulty in making one; the difficulty was to make a turbine which would produce any useful work at the engine shaft, and for a reason which be would very soon make clear. In the Otto cycle engine the compression was produced almost entirely in the motor cylinder itself, and therefore any heat which escaped from the charge of air and gas into the walls of the cylinder was only there for a short time, if at all. When the charge exploded, the cylinder walls were heated, and cooled on exhaust, but at any rate they were probably very much above the temperature of the compression charge. It was clear, he thought, that no turbine could ever be made to work on that principle. The compression of the air and gas must take place outside the turbine casing. Therefore one had to contemplate the possibility of economically producing that compression.

The author had alluded to the rotary compressor. The rotary compressor, as far as he knew it—and that was only up to about 7 pounds or 8 pounds per square inch—was singularly inefficient, and he felt convinced that any rotary compressor at present known would take the whole power of the engine to compress the air and gas to begin with, because its efficiency was probably not more than 0.4; and, when it was considered that the negative work was 0.4, it would be found there was about 30 per cent. of the work of the engine absorbed in compression. Referring to the experiment of Diesel, by taking a portion of the air during compression and compressing it in an independent cylinder, a very much higher efficiency had been obtained. On that point he could answer the question with regard to the efficiency which Diesel got in his compression cylinder. The pressures were not correctly stated in the Paper, though perhaps they were the pressures found by Mr. Ade Clark. In his own Diesel engine the compression in the cylinder was 35 atmospheres, and the compression on the blast used for spraying the oil was 57 atmospheres. The efficiency in the small compressor was not high, but then that small compressor took such a very small charge that it did not affect the general efficiency of the Diesel engine, or hardly appreciably.

When dealing with the steam engine, one was dealing with a fluid that could be carried over very large distances without very much loss of heat. A 5 per cent. liquefaction in a pound of dry steam would liberate quite a considerable quantity of heat. In the gas engine, however, those conditions were reversed. Any fall of temperature of the gas came entirely from the internal energy of the gas, and therefore an explosion charge could not be trusted to remain in contact with the metallic surface for any length of time without an excessive loss of heat. From that, he inferred that any form of Parsons turbine was not the most desirable form for working with gases. The cooling in passing through the comparatively small passages would be so immensely rapid as to prevent anything like a reasonable efficiency being obtained. In the reciprocating engine there was a rather different set of conditions. There was a large volume of gas in a cylindrical vessel, and only the outer portion of it became cooled at all, the centre core remaining always hot, there being not sufficient time to pass its heat to the outer walls. Hence the losses from cooling in a reciprocating engine were not serious.

With regard to the possibility—and he was afraid it was the only possibility in the last instance—of using the divergent nozzle, the author had clearly indicated what were the limiting conditions for that divergent nozzle, namely, that a velocity was required of about 5200 feet per second in order to turn the heat energy of the charge into the kinetic energy of the turbine vanes. Probably, at present, that was not possible, but he saw no reason whatever to suppose that, with the advances in metallurgical science, engineers would not be in possession of a material which would stand the high stresses and which might probably be lighter. It seemed to him by no means impossible to get that, and, granting the fact of a material, then the turbine was materialized almost at once. It was a matter of mechanical difficulties. As he had stated before, there was no reason to suppose that it would be anything like economical, owing to the very high ratio of the negative work, which did not occur with the reciprocating engine.

The questions which the author raised with regard to data were of course extremely important, particularly the questions relating to the reciprocating compressor and the expansion of hot gases in the divergent nozzle. The author would probably know very well that any one of those experiments was a most serious matter to undertake, and to get results which were in any way commensurate with practical work meant conducting the experiments on a large scale. When experiments such as those were conducted on a large scale the expenses were apt to be enormous, and therefore one could hardly expect any private individual to give his information for the good of the world at large. If the experiments were made, no doubt they would be made by some individual who, naturally, was seeking his own advancement and who wished to produce a commercial article. Whether such a thing was possible he did not know, but he felt very strongly there was no possibility of advance in that direction until more efficient compressors were obtained. That was the real gist of the matter. In the last five years he had had on the average from six to eight patents for gas turbines put before him, and in every case he had discovered that they were perfectly vague on the subject of the negative work. If the author only succeeded in persuading engineers that that was the real rock on which they split in devising a gas turbine, then he thought the object of the Paper would be thoroughly accomplished. He thanked the author for having brought such a subject before the Institution, even if it were only in the form of a scientific investigation.

Mr. James Atkinson said the author had given a most excellent Paper on the theory of gas turbines, and as a matter of fact the gas turbine simply existed now in theory. Whether it would remain so or not was a question that might take a good deal of deciding. With regard to the temperature which the author assumed could be put into a turbine of the Parsons type, the author had taken it as 700° C. (1292° F.). Professor Burstall had spoken about the probabilities of a metal which would stand those temperatures and the scouring action which must take place in a turbine, but he, Mr. Atkinson, thought such a metal was a long way from being discovered. Any metal that had iron in its composition commenced to oxidize at about 400° C. (752° F.), or 500° less than the temperature given by the author. If that temperature existed in the presence of free oxygen, metal would begin to oxidize, and the scouring action which took place in the turbine would very soon wash away the blades. He thought the author might have made a slight mistake in talking about superheated steam in steam-engines and the efficiency being only a small percentage. As a matter of fact, he thought superheating in the steam engine was more efficient than the actual steam working in the steam engine. In a turbine, superheating the steam increased the efficiency very largely, and it would increase the efficiency of a steam engine equally as well, if it were not for the fact that superheated steam went a very small way through a steam engine, and in a triple-expansion engine it very rarely went beyond the high-pressure cylinder, if as far as that. He expressed his admiration for the care, attention, and trouble the author had taken in producing his calculations, which would be of very great use to engineers who had to deal with the question. The Paper practically did for the gas turbine what Mr. Dugald Clerk did for the gas engine many years ago.

Lt.-Colonel R. E. B. Crompton, C.B., thought that the two great difficulties which stood in the way of producing a gas turbine had been fully stated by Mr. Davey and Professor Burstall, the latter of whom had rounded up the subject so completely that it left little for others to say. There was one point, however, which might perhaps be mentioned, although he was uncertain whether it could be fairly discussed under the question of gas turbines. He had recently been present at the Electrical Congress at St. Louis, and had taken part in the discussions which followed two interesting Papers, read by Mr. Emmett, of the General Electric Co., of Schenectady, on the Curtis form of steam turbine, and by Mr. Hodgkinson, of the Westinghouse Co., of Pittsburg, on the Parsons form of steam turbine. Several speakers appeared to think that one method of obtaining the maximum efficiency in electrical energy from the energy in the coal would be by using the gas producer and gas engine to deal with the higher temperatures, and by utilizing the energy remaining in the jacket-cooling water and in the exhaust of the gas engine by raising steam in a suitable boiler and utilizing this steam in a low-pressure steam turbine. In this way it seemed possible to increase the present highest practical efficiency obtainable with the gas engine from the figure of about 28 per cent. up to possibly 38 per cent. He thought that there were no practical difficulties in the way of the mechanical engineer in carrying out this development. What would then remain to be considered would be whether the economical advantage in increased efficiency would not be more than counterbalanced by the extra first cost and afterwards by the maintenance cost of the extra plant that would have to be added to obtain this increased duty from the fuel. It appeared to him that, in many cases where fuel was dear, it was quite probable that the answer would be favorable, and that therefore such a proposition ought to be seriously taken in hand by those interested in the question. Speaking as a power engineer, he thought that this combined use of gas engines and steam turbines was likely to be of more immediate practical importance than the remote possibility that the mechanical difficulties connected with the gas turbine would be successfully surmounted.

Mr. H. M. Martin said that Professor Burstall had mentioned, in connection with rotary air compressors, 0.4 as the highest efficiency he had come across. In a turbine compressor built for the Farnley Ironworks, in which the rotary air compressor was driven by a steam turbine, the combined efficiency as determined by Professor Goodman was 61 per cent. That was to say, the "air horse-power" was 61 per cent. of that theoretically due from the steam. This meant that the efficiency of the compressor *per se* was something like 80 per cent.

With respect to experiments on divergent nozzles, it might be stated that some had been made by Professor Stodola, who found that with a pressure drop of from $10\frac{1}{2}$ to $\frac{1}{10}$ atmospheres, the loss was 20 per cent. The experiments were not entirely satisfactory, as the measurements were made by means of a small tube passing centrally down the nozzle, which acted as a sort of Pitot gauge. In fact, Professor Stodola concluded that in the absence of the resistance caused by this tube the loss would have been about 15 per cent.*

* The use of a thermo-couple would seem to afford a means of measuring the loss in a divergent nozzle with the least possible disturbance of the flow. With such an instrument the distribution of temperature throughout the nozzle could be determined, and this known, together with the weight discharged per second, the efficiency of the nozzle could be readily calculated by elementary thermodynamics.

There was another possibility with respect to gas turbines which he did not know whether it would prove practicable to work out. Consider the well-known water ejector, such as indicated in Fig. 34. High-pressure water entering through a suitable nozzle into a combining cone drew in and

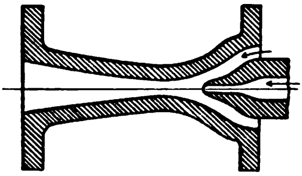


Fig. 34.—Water ejector.

carried with it low-pressure water to form a combined jet. The efficiency claimed for the appliance was over 90 per cent. He did not know whether it would be possible to work an air ejector on the same lines. The method would be to let a jet of products of combustion at high pressure develop

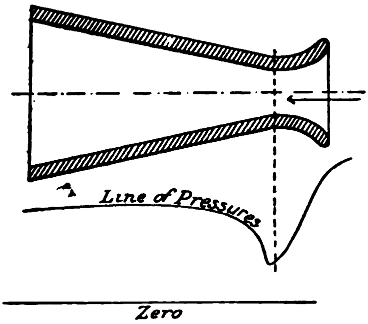


Fig. 35.—Divergent nozzle, showing varying pressure.

its full kinetic energy in a divergent nozzle. This would then be utilized to draw in, say, four or five times its weight of air by an ejector, and the combined jet having a correspondingly reduced velocity and temperature would be utilized to drive the turbine. Some experiments by Professor Stodola went to show that the necessary suction could be obtained with an air-jet. Thus he found that if a divergent nozzle designed for a high drop of pressure was used for a smaller drop, the distribution of pressures in the nozzle was such as indicated in Fig. 35 (page 89). Obviously, if holes were put through the walls of the nozzle in the region of low pressure, air would be drawn in, and Professor Stodola had attempted to construct an ejector somewhat on these lines, but so far had failed to obtain an efficient apparatus.

Professor ROBERT H. SMITH said that much useful and interesting information, especially upon the outflow of high-pressure fluids through varying shapes of nozzles, was to be found in the book by Professor Stodola on "Steam Turbines."* As the author in his Paper referred to a theory that more than a certain critical velocity could not be obtained in such nozzles, he might mention that Professor Stodola's experiments showed velocities in the nozzles which were considerably higher than the critical velocity calculated on the assumption which the author referred to; but when the critical velocity was passed, extremely interesting phenomena occurred, which had been suggested by the diagram shown by Mr. Martin, Fig. 35 (page 89). The pressure oscillated, not in time, but in distance along the axis of the nozzle. Great waves, steady waves, of pressure existed. The most interesting practical point resulting from these experiments was that such waves of pressure did not serve any useful purpose, and that the nozzle should be designed so that they did not arise. The nozzle could be designed for each discharge pressure so that the waves could be

* "Die Dampfturbinen," by Dr. A. Stodola, of Zurich, 1903, published by J. Springer. Berlin. *See* pages 17–37. English translation by Dr. L. C. Lowenstein, published by D. van Nostrand Co., New York, and Archibald Constable & Co., London.

Other recently published books on the subject are "Le Turbine à Vapore et à Gaz," by Ing. Guiseppe Belluzzo, published by U. Hoepli, Milan, 1905; and "Dampfturbinen, Entwickelung, Systeme, Bau u. Verwendung," by Wilhelm Gentsch, published by Williams and Norgate, London, 1905.

smoothed down and made to disappear. That, he thought, was the most useful result of Professor Stodola's very long and exceedingly interesting series of experiments last year.

The difficulty to which Mr. Atkinson had alluded seemed to be the greatest difficulty to be feared in gas turbines, because the fluid resulting from combustion must be diluted very greatly in order to keep down pressure and temperature within practical limits, and the only cheap way of diluting it was with an excess of air. There would be, therefore, a very large excess of extremely hot air containing large quantities of free oxygen passing through the narrow nozzles and wheel buckets, and he fancied these would be rapidly burned away. He was surprised to hear the statement that air compressing could not be done at a greater efficiency than 40 per cent. He knew many reciprocating air compressors in which it was possible to easily attain over 60 per cent. efficiency. Mr. Henry Davey had sketched an apparatus he had made upon a small scale, Fig. 33 (page 80). He, Professor Smith, had spent two years in experimenting upon a similar apparatus on a larger scale, which was designed for 600 pounds working pressure and 45 indicated horse-power, and he had worked it at 130 pounds; but it was not for the combustion of gas, but for that of oil. The difficulty had been, so far, one similar to that which Mr. Davey referred to, namely, the occasional sudden extinction of the flame and the difficulty of controlling the flame, leading, in his apparatus, to an inconvenient rush of the water over into the fire-tubes.

In the Paper there was a very interesting and important remark (page 29) relating to Carnot's formula for the efficiency of an ideal heat engine. This formula was not intended for practical application to working plant, but was framed only for the purpose of proving a theoretical proposition, a proposition of immense value in the science of thermodynamics. It was not a measure of efficiency applicable to any heat engine which it was possible to construct and work. The upper and lower limits of temperature might be greatly changed without to any extent changing the thermodynamic efficiency. It was an historical fact that every engine that had been made and worked successfully had cut down its upper temperature limit before it had obtained practical success.

He was struck by the remark (page 76) that "With the average user of a small engine, producing, say, 100 B.H.P.-hours per week, a reduction of £10 in the initial cost is of more consequence than a reduction of 2 cubic feet per B.H.P.-hour in gas consumption." It occurred to him that that was rather an extravagant estimate of the value of low initial cost. It was usually his fate to argue in favor of greater consideration of initial cost. Considering that 2 cubic feet per B.H.P. at 100 B.H.P.-hours per week was 200 cubic feet of gas per week, and taking gas at 2*s.* 6*d.* per thousand, that meant 26*s.* a year; 26*s.* was 13 per cent. upon £10, and 13 per cent. was rather a high percentage to take in the comparison. Perhaps a saving of 3 cubic feet of gas per B.H.P.-hour in an engine of this small size working with so bad a load factor might be more fairly comparable with the advantage of £10 decrease in capital cost. On page 76 the author suggested the use of gas turbines upon motors for certain reasons which he gave, but surely the difficulty of using any sort of gas engine on the motor was not in the engine itself but in the weight and bulk of the vessels that had to be carried to hold the gas. Some years ago he worked that out, and he found it was practically impossible, even with the high pressures that were used in weldless steel gas cylinders, to carry any reasonable bulk of gas in a motor car. Lower down on the same page the author made a very useful remark, pointing out that in order to displace electric driving the turbine would only require to possess an efficiency greater than the efficiency of the central station engine multiplied by the efficiencies of dynamo, mains, and motor. If those four efficiencies were multiplied together a very low compound efficiency would be obtained in the end.

Mr. Neilson, in reply to the discussion, stated that Professor Burstall's remarks on rotary compressors and their efficiencies had been already replied to by Mr. Martin (page 88). It by no means followed that an efficient rotary compressor could not be used, although the turbine compressors were not successful at high pressures. Turbine compressors had been used up to about 80 pounds pressure or so, that was, compressors arranged like a converse steam turbine. It was quite reasonable to suppose that such compressors would not be efficient at high pressures, as they had a great number of stages, and the air was somewhat roughly handled at each stage. He thought there was great room for investigation in the matter of trying to find an efficient rotary compressor on another principle. Professor Burstall had stated that a gaseous charge could not be allowed to remain in contact with the metal for long without a great transference of heat, and that gas was very different from steam in that respect. In steam turbines with superheated steam there was practically a gas at the high pressure end of the turbine. Superheating had very much improved the efficiency of steam turbines, and it would be reasonable to suppose that no great loss of heat occurred at the high pressure end of the turbine. He believed experiments had been made at the Manchester Technical School with a Parsons turbine, and the temperatures taken at different stages of the expansion, but he had not seen the figures. Those experiments were with steam slightly superheated.*

Professor Burstall had referred to the possibility of getting a material some day that would stand the high stresses caused by high speeds. During the last few years there had been very great improvement in that respect. He did not think that a De Laval turbine, such as the 300-H.P. now

* The author subsequently saw the figures through the courtesy of Mr. Mellanby; but the superheat was so slight that he considered the figures, although interesting, were of no value in the present instance.

built, could have been built 20 years ago, as there was no material then to stand the enormous stresses. It had to be remembered that the stresses varied pretty nearly as the square of the velocity, so that, if the velocity was doubled there was four times the stress. The experiments mentioned at the end of the Paper certainly would cost a great deal of money. Those with divergent nozzles would perhaps entail least expense, as such experiments would not require a great amount of apparatus. Professor Stodola's experiments were exceedingly interesting, but it was impossible to take the results as being very exact. Professor Stodola admitted that his measuring apparatus was of such a nature as would be apt to distort the results. For example, in his nozzle he had a measuring tube which was very small, but still of a diameter that was appreciable considering the minimum diameter of the nozzle. Experiments on the effect of wind pressure on different bodies had been made and they were described before the Institution of Civil Engineers* by Dr. T. E. Stanton. The experimenters discovered that in order to get reliable results the models they put in the air duct had to be very small compared with the dimensions of the duct. That was with a comparatively small velocity of air. In Professor Stodola's experiments the velocity of the fluid was enormous, and the effect of any body in the centre of the jet would presumably make a much greater difference. Therefore, while Professor Stodola's experiments were interesting, further investigations were required in connection with the matter.

With regard to the fluctuation of pressure mentioned by Mr. Martin and Professor Smith as being found by Professor Stodola, the steam in passing through certain nozzles fell in pressure more than it should do, and then fluctuated up and down in regular beats before it remained steady at the final pressure. Fig. 36 illustrated the oscillations of pressure, which were like those of a spring. In the diagram, heights

* Proceedings Institution of Civil Engineers, 1903, Vol. clvi, page 78.

represented pressure, and horizontal distances represented measurements along the nozzle. He did not think Professor Stodola was the first to discover that. It illustrated what he had said earlier, that it was very difficult in many cases to tell to whom credit was due. The fact was mentioned in a Paper* which Mr. Hodgkinson read in America some few years ago, but even he did not seem to be the first to discover the phenomenon.

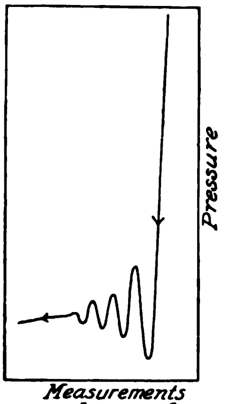


FIG. 36.—Fluctuating pressure of steam passing through nozzle.

Mr. Atkinson had referred to the statement about superheating (page 30). Of course, superheating did increase the efficiency, *not owing to thermodynamic reasons*, but because of the reduction in friction and for other reasons which he need not go further into. Mr. Martin alluded to the proposal to draw in an extra quantity of air to mix with the products of combustion in order to cool down the temperature of the explosion and get a more reasonable temperature, and he mentioned the proportion as 4 pounds of air to 1 pound of products of combustion. The lower temperature suited the turbine, but it had to be remembered that in a practical turbine the extraction of the available energy of the

* Proceedings, Engineers' Society of Western Pennsylvania, Nov., 1900.

fluid was never complete, and that with this air-dilution scheme, instead of discharging 1 pound of fluid with a certain amount of unrecovered energy, it would be necessary to discharge 5 pounds, thus throwing away five times the available energy that would be otherwise thrown away. If water instead of air were used to dilute the products of combustion, the mass of fluid would not be increased to the same extent. A somewhat similar proposal had been made as regards steam turbines, namely, to dilute the steam from a divergent nozzle with water or other fluid in order to bring down the velocity so as to enable the velocity to be utilized conveniently in the turbine wheel. This would allow the wheel to run at a lower speed. Instead, however, of discharging 1 pound of fluid with a certain final velocity, the scheme would involve the discharging of, say, 5 pounds or 6 pounds.

Professor Smith had referred to the question of the relative advantage of low initial cost and low working expenses (page 92). The case mentioned by Professor Smith he had worked out, but he forgot what the saving amounted to as a percentage of the extra capital. Taking it as 13 per cent. (as given by Professor Smith), he thought that with a small turbine the reduction in initial cost would be of more importance than the 13 per cent. saving. It was not like putting money into a building. No one could say how long the engine was going to be up-to-date, and no one knew how long it would be before another engine was required. Moreover, small users generally wanted money at the time. He thanked the Institution for the vote of thanks that had been accorded to him.

Communications.

Mr. Edward Butler wrote that perhaps the greatest obstacle in the way of the production of an efficient gas turbine—namely, "the very low reactionary value and destructive property of a highly heated gas, as compared with steam of the same pressure"—could be removed by a combination of apparatus in which gas and air would be supplied to a mixing chamber communicating with a series of long explosion chambers with siphon formations for receiving water. The *modus operandi* being for the series of explosion chambers to be charged with explosive mixture and fired in rotation, the effect of the explosions would force out the water contained in the well or siphon formation of each tube in succession and this water could be directed on to the wheel of a water turbine. Succeeding charges of gas mixture would enter automatically from the mixing chamber through non-return valves immediately after each explosion. The only operating mechanism required would be means for quickly filling the siphons with water and for igniting the mixture in the series of five or six explosion tubes, by which an almost continuous stream of water would be directed onto the turbine wheel at high velocity, thus avoiding all the trouble that would follow from any method of employing a turbine wheel actuated direct by the hot rarefied gases of combustion. He offered this solution of a gas power turbine, after having had some experience in the working of a steam turbine worked somewhat on these lines, namely, with water accelerated by a steam injector and boiler.

Mr. Dugald Clerk wrote that he had read the Paper with much interest, and he quite sympathized with the author in his effort to clear up in a preliminary way the many abstract points which required consideration before the practical problem of the gas turbine could be attacked with any chance of success. In view of the wonderful results obtained by the Hon. C. A. Parsons and his imitators with the steam turbine, it was very natural that the attention of engineers should be called to the problem of the gas turbine. Belonging to the older school of engineer inventors and having become somewhat mentally fatigued by the numerous difficulties experienced with cylinder and piston gas engines in every stage of their progress, it might be that the writer was less likely to take a hopeful view of the gas turbine problem than a younger engineer, whose life and practical engineering difficulties were still before him. Whether this were so or not, he must confess that his view of the future of the gas turbine was not so hopeful as the author's. The difficulties appeared to be even greater than the author had apprehended. Mr. Neilson rightly laid a certain stress on the relatively low efficiencies possible in turbine compressing plants; but he did not think he had laid sufficient stress upon the similarly low efficiency of expansion in steam turbines of existing types. No doubt the author had clearly in his mind the fact that there were special advantages in the steam turbine which counterbalanced the disadvantages of relatively inefficient expansion. The steam turbine had, to begin with, the great advantage of practical freedom from losses due to initial condensation. It had the further great advantage of expansion to a much lower pressure point than could ever be effectively attained with a reciprocating steam engine. These two advantages, in the case of steam, undoubtedly enabled the Parsons turbine to reach figures of economical steam consumption which were not touched at all by ordinary reciprocating steam engines, and were very rarely reached even in special reciprocating steam engines using high superheats. These advantages of no initial condensation and largely extended expansion were advantages special to steam; they were not advantages which would be found in using a turbine construction even for the purpose of expanding high pressure cool gases, such as compressed atmospheric air. In an ordinary gas engine cylinder, the efficiency of compression and expansion of the gaseous contents of the cylinder (without combustion at all) was so high that practically no difference could be detected between the compression and expansion lines of the air within the cylinder during successive compression and expansion strokes. Practically, what one might call the efficiency of gaseous compression and expansion in an ordinary gas engine cylinder was certainly not less than 99 per cent. This, undoubtedly, enabled engines to be operated economically with large proportionate values of negative work. If, however, the efficiency of compression had been, say, 70 per cent.—a figure higher than any turbine air-compressor could give at present—and the efficiency of expansion 80 per cent.—a figure higher than any existing turbine could give operated by expanded compressed air—even then the united efficiencies would only amount to 56 per cent. That is, assuming the gases to be heated for the expansion period, the loss to be made good to bring the diagram up to unity would be 44 per cent.; that is, if the volume of the gas were increased by heating from 56 to 100, then the expansion would only be sufficient to produce a diagram which would keep the engine running if it were quite frictionless. In the writer's view, therefore, the problem to be faced required not only the invention of a turbine air compressor compressing, say, to 200 pounds per square inch, with an efficiency of something nearly 90 per cent., but it also required the invention of a turbine motor engine which would give a like efficiency of expansion of the gases so compressed. High efficiencies of compression and expansion of gases, such as air, were undoubtedly theoretically possible; but he knew of no turbine yet in existence which would give any efficiencies such as he had indicated. Until these efficiencies were obtained, it seemed to the writer impossible to design any gas turbine having a chance of success.

The author had very accurately made clear the point that, in some types of turbine, low temperature, and therefore large negative work, was necessary for working conditions. He himself could not help thinking that, although the high temperatures he proposed—2000° C. (3632° F.)

and so forth—certainly diminished negative work, they yet introduced for the inventor a much worse condition from the points of view of durability of blades, and heat losses if the blades were kept cool. There seemed to be no possibility whatever of working a turbine at temperatures approaching 2000° C. The De Laval type was proposed by the author to enable temperatures to be reduced while the energy was maintained in the shape of velocity of the moving gases. He could not help thinking that any scheme which allowed of such reduction would also cool the expanding gases by the very conditions required in the constructions of the expanding nozzle. He did not see any immediate future for a gas turbine, except in possibly utilizing the exhaust gases from reciprocating engines, which at present were liberated under considerable pressures. He feared that the attack of the problem of an efficient turbine air expander was more within the province of the practical inventor than the scientific investigator. He would be much interested to hear any ideas Mr. Neilson might have in the direction of producing efficient turbine compressors and expanders.

He hoped that the author would not be discouraged at the somewhat pessimistic tone of these remarks. He had often thought over this subject, and had not as yet seen any hope of getting as good results, either as to economy or durability, with gas turbines as were obtained with reciprocating engines.

Mr. W. J. A. London wrote that, in studying the Paper, one could not help feeling impressed by the thoroughness with which the author had dealt with the various possible cycles to which engineers were to look in the future gas turbine. The most interesting point, however, in the writer's opinion, was the way in which the author's investigations showed the difficulties to be met with in the successful designing of such machines. It was a pity that he had not attempted to consider from a more practical standpoint several of the cycles set forth; had he done this, the writer could not help feeling confident that he would have omitted several of the cycles referred to.

On page 32 a reference was made to a combustion chamber where the burning gases could rest before entering the turbine. He thought this would be a great source of loss, because this chamber would have to be cooled, and, to allow the gases to remain in a chamber with cooled walls, a great amount of efficiency would naturally be lost. Further, on the same page, the author referred to a mixture of air and fuel being always supplied to the flame with a velocity greater than that of the propagation of the flame. This, in the writer's opinion, would be a very difficult thing to do, especially at the high outgoing velocities attending the expanding gas. On page 39 the author referred to the Parsons turbine with steel blades for working at about 700° C. (1292° F.). This temperature was very high, and it was doubtful whether any blades with sharp edges could be made to withstand it for any length of time.

In Cycle I, Case 2 (page 39), the author proposed to circulate water in the revolving part. If he would consider the difficulties to be met with in circulating water in the revolving chamber, the writer thought he would find that it would be a more difficult problem than he considered. The water would be thrown against the outer walls, which was the point to be desired, but whether the circulation could be arranged without seriously interfering with the running balance of the machine was another question. Further, even if the rotating drum were cool, this would not effectively cool the blades, and to do this as the author suggested, by making the blades hollow, would require very large blades with internal pipe arrangements for ensuring circulation. With his device he claimed that 2000° C. was allowable; this corresponded to 3632° F., or above the melting-point of steel, so that it was hardly likely that any sharp edges could be expected to remain on the blades. For mechanical reasons he did not think one should look for the gas turbine on the Parsons principle, but more on the lines of the De Laval, as set forth in Cycle I, Case 3a (page 48), where the gas was expanded in nozzles, and the temperature reduced before the working fluid came into contact with the moving blades.

The author referred (page 56) to a combination of reciprocating engines and turbines working with gas on the principle now being adopted in connection with steam engines and turbines. It would be interesting to learn what gain this system had over reciprocating engines. The idea in the steam combined system was that the reciprocating engines were perhaps more economical at lower speeds than the normal. The steam turbine working non-condensing had great difficulty in competing with the reciprocating engine, but when working condensing the conditions were different, and the turbine was more capable of taking care of what might be called the tail end of the expansion curve. These conditions when working condensing did not come into account when working non-condensing, as in the case of a gas turbine, so that it was doubtful whether the combined system would be even as economical as a gas engine of the ordinary reciprocating type.

The system which the writer thought offered the greatest possibilities was on the principle of Cycle III, Case 1 (page 59), where the steam jet was inserted into the gas, thus utilizing the expansion of the superheated steam and reducing the temperature of the working fluid to within practical limits. The ideal efficiency of 0.33 and the ratio of negative work of gross work of 0.41 did not look as satisfactory as some of the other cases set forth, but it was undoubtedly more practical, and it seemed more than likely that the solution of the difficulty would be found in this direction. Referring to the author's remarks (page 69) on the velocity of gases issuing from diverging nozzles, there must be a limiting velocity for the flow of gas through an orifice, this velocity being at a point where the weight discharged multiplied by the velocity due to difference in pressure was a maximum. He thought this point had been experimentally investigated, and it had been found that the point of maximum discharge was when the ratio of internal to external pressure varied between 0.5 and 0.6 according to the nature of the gas. This pressure of 0.5 to 0.6 initial pressure would be at the throat, or, in other words, at the smallest part of the nozzle, and the velocity had been found to be somewhere about 1500 feet per second. However much the difference of pressure was increased, this law remained good, and no more gas would pass through the orifice, nor would the velocity at the throat increase. The terminal velocity, however, would be greater in proportion to the drop in pressure.

Mr. E. Kilburn Scott wrote that he thought the author struck the keynote of the present position when he suggested (page 78) that a thorough set of experiments should be made and the results published. The information so obtained by the engineering world would be extremely valuable, whatever the price, as it would save much time and overlapping if the several fundamental considerations involved could be investigated straight away. It was essentially a matter for some public body, and he suggested that the Institution of Mechanical Engineers should undertake the work. The tendency of the times was well shown by the working arrangement between the General Electric Co. of America and the Allgemeine Elektricitäts Gesellschaft of Berlin, whereby experimental data, drawings, etc., were now the common property of both firms. This arrangement was come to with the idea of cutting down expenses, and at the same time facilitating the progress of experimental research and of new designs.

The high temperatures and high velocities involved in working turbines with gas, and the fact that the hot gases came so intimately into contact with the metal parts, called for an investigation as to the most suitable metal to employ.

Separate blades, caulked in as with the Parsons turbine, would hardly do for gas turbines, as this construction was not even satisfactory at the temperature of highly superheated

PRINCIPLES OF TWO GERMAN ARRANGEMENTS OF TURBINES.

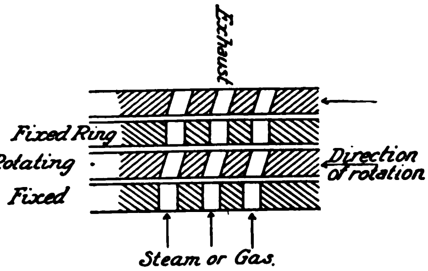


FIG. 37.—Principle of Bucholz turbine.

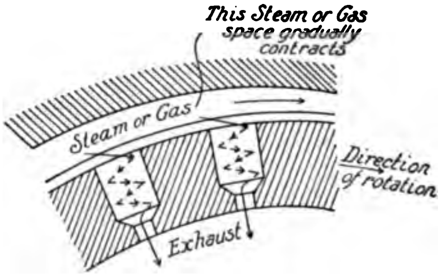


FIG. 38.—Principle of Patschke turbine.

steam. New turbines were, however, coming forward which had not so delicate a constitution. The Bucholz,* for example (Fig. 37) had a series of metal discs with holes drilled through them at an angle, and against which the working medium impinged; and the ingenious Patschke turbine (Fig. 38) had the rotating part simply in the form of a drum with a number of holes drilled in the rim. With such

* "Electrical Review," January 8th, 1904, page 66.

solid construction, exceedingly high velocities could be attained, and temperature had very little effect, whilst these two designs had the further advantage of being reversible.

The question of water-supply to the heated metal was mainly a matter of mechanical design, and he thought it was less difficult to arrange water-circulation in the interior of the rotating part of a turbine than it was to get it into the pistons and exhaust valves of a large slow-running power gas engine.

In comparing gas turbine with electric motor driving, it must be remembered that practically all turbines ran at speeds which were unsuitable for driving shafting and machine tools, and that the latter were *par excellence* the very things for which the electric motor was most suitable, by reason of the ease with which the motor could be started and stopped or its speed varied, and also moved about for use with portable tools. A pipe containing gas could never compete with an electric cable for such work. As a matter of fact, the author need not worry as to uses for gas turbines; only let it be clearly demonstrated that they were commercial, and they would become the favorite prime-mover for driving electric generators, and in that direction alone there was scope enough. Any one prophesying two years ago that the Curtis turbine would attain its present position in the United States would have been thought a dreamer, yet the writer had recently seen five stations equipped with 8000 horse-power Curtis sets running without a hitch. So far as power station work in the United States was concerned, steam turbines were preferred to reciprocating steam engines, and he believed that one day gas turbines would come to the front just as rapidly.

Mr. George A. Wigley wrote that he thought it was remarkable that during the discussion no direct reference was made to oil turbines or engines, with the exception of a few remarks on the Diesel engine. One of the first problems to be considered with regard to a gas turbine was the production of the gas, and if it were necessary to supply a gas-producer even of the modern suction type, a pump for compressing this gas and a mixing chamber, it was useless to compare such a plant with an electric motor, irrespective of maintenance costs, but if a mineral oil were substituted for the gas the problem immediately assumed a different aspect.

In Cycle III, Case 2 (page 61), the author clearly pointed out the advantages to be obtained by the use of water or steam with the gas, and this would, of course, hold good with oil as fuel, the ratio of negative work to gross work being decreased, but it was this question of negative work which, in the writer's opinion, was most important. In oil (mineral oil) they had what was practically a gas compressed to liquefaction all ready to hand in a very convenient form, and in such a condition that, when intimately mixed with the proper amount of oxygen, it formed a gas which was ready to give up a great proportion of the work which would be or had been necessary to compress it to liquefaction. So that they had what was to all intents and purposes a gas already compressed; and as this gas had had work done upon it far greater in amount than was necessary for the required immediate object, it followed that the oil, when fired, gave up this work, and not only reduced the negative work given out by the turbine to the extent of that required to compress the gas to the necessary pressure, but also reduced the excess work required to liquefy it, less that corresponding to the latent heat of vaporization. The result was that the negative work calculated for gas was greatly reduced if oil were used, the extra work being given in the convenient form of heat and kinetic energy of the particles.

Mr. Neilson, in reply to the written communications, wrote that Mr. London had raised the question as to whether if the burning gases were allowed to rest a short interval in a combustion chamber before being taken to the turbine, there would not be a great loss of heat. He (the author) agreed that this question deserved consideration, but he did not think that there need be anything like as great a loss of heat to the walls of this combustion chamber as to the walls of the combustion chamber of a reciprocating engine, which required to be kept much cooler than would be necessary in the case of a turbine. As regards the last remarks made by Mr. London, it must be noted that the highest velocity obtainable when a gas issued from a simple orifice was not necessarily the highest velocity that could be obtained when a divergent nozzle was placed on the delivery side of the orifice.

With regard to supplying the air and gas at a velocity higher than the velocity of the propagation of the flame, mentioned by Mr. London, he did not think that this was really a difficult problem with the gas turbine. With a reciprocating engine it was complicated, but with a gas turbine having a continuous flow of gas it should not be difficult. With a mixture of a given gas and air in constant proportion, the flame would have a given rate of propagation; and, if care were taken to supply the air and gas to the combustion chamber at a greater velocity, there should be no firing back. Mr. London had referred to the proposal to circulate water in the turbine. It was said in the Paper that this had been proposed, but he (the author) did not propose it himself, and he had not much faith in it.

He agreed with Mr. Kilburn Scott as to the benefit to be derived from experiments made by a public body.

CHAPTER III.

THE DISCUSSION BEFORE THE SOCIETY OF CIVIL ENGINEERS OF FRANCE.

ONE of the most complete papers upon the subject of the gas turbine which have yet appeared was presented before the Society of Civil Engineers of France, on February 2, 1906, by M. L. Sekutowicz, this covering a thermodynamic treatment of the question as well as a study of the gas turbine from a mechanical point of view. This paper elicited an animated discussion which included the views of a number of eminent specialists who had devoted attention to the subject, the whole appearing in the Mémoires of the Society for February 1906.

After a brief historical introduction, M. Sekutowicz proceeds with his thermodynamic study as follows: *

As is the case with all heat motors, the gas turbine involves the action of a given mass of a determinate fluid, during a unit of time, between certain limits of pressure and temperature.

In order that this action may be subjected to computation it is necessary that each of its phases be considered as simple; that is, characterized by the constancy of some one of the independent variables upon which the state of the fluid depends at each instant, such as: volume, pressure, temperature, and entropy.

The phase of compression may be adiabatic or isothermic. The introduction of the heat may he isothermic (as in the Carnot cycle and its derivatives), or isobaric (combustion under constant pressure), or isopleric (combustion under constant volume: explosion). The expansion will be adiabatic. Finally the rejection of the heat at the lower tem-

* Mem. Soc. Ing. Civ. de France, Feb. 1906, p. 204 et seq.

perature, can hardly be other than isobaric or isothermic (the Carnot cycle).

The first two phases are the most important, for the last two can be only partially realized in practice.

In fact the expansion, in the case of the gas turbine, will be almost rigorously adiabatic, while the rejection of heat can only take place at constant pressure to a regenerator; or, what amounts to the same thing as regards the cycle, by the non-closure of the cycle.

We then have six principal variants. Besides these we may examine the cycles of Stirling and of Ericsson, in which the adiabatics of Carnot are replaced by the isodiabatics and the regenerative cycles derived therefrom. Finally we have to study the effects of the injection of water or of cold gases.

Heat motors are too frequently studied with regard to the extreme temperature limits of the cycle. There results a misleading comparison with the Carnot cycle, or at least a comparison which fails to indicate the actual development. What really concerns us is the value of the thermal efficiency ρ, the mechanical efficiency η, and the total useful effect represented by their product.

Now, as we shall see, these efficiencies are not always limited by the extreme limits of temperature. Other factors, no less important, must be considered. In the present case these are: the limits of pressure, the ratio of the work of compression to the useful work, and the quantity of useful work produced by a kilogramme of air.

The lower temperature limit is usually that of the atmosphere, and may be taken as about 300° C. absolute. The final temperature of the expansion, however, will generally be much higher. This value is most important, since it is the temperature of the gases delivered upon the rotating metallic portion of the turbine.

We will assume that the turbine wheel can be so constructed as to stand a temperature of 700° C. absolute, without injury. This fact has been fully demonstrated in practice. In all that follows, therefore, we shall consider the temperature at the end of the expansion as being 700°, except when examining the influence of variations of this factor.

The upper temperature limit is determined largely by the heat resistance of the refractory material used, not only for the lining of the combustion chamber, but also for the construction of the nozzle in which the expansion takes place. There are now available such substances as carborundum, which are capable of resisting the highest temperatures developed.

Under these circumstances the maximum temperature is limited by the following conditions:

1. It must be such that, with the degree of expansion attainable, the final temperature of expansion shall not exceed 700° C. absolute.

2. It must be attainable by the combustion of ordinary fuels with a sufficient quantity of air to insure a complete combustion.

As is well known, combustion under constant volume produces a much greater elevation of temperature than is caused by combustion at constant pressure. Besides this, when the compression is adiabatic the temperature of the gas is raised to a greater or less degree before combustion, this effect being added to the temperature of combustion. For a compression of 30 atmospheres the temperature will reach 800° C. absolute.

Thus, illuminating gas requires 5.5 times its volume of air in order to enable perfect combustion to be effected. If, in practice, we assume that 6 volumes of air are required, 1 kilogramme of the mixture will evolve 574 calories. Under these conditions, starting from the ordinary atmospheric temperature, the combustion, if conducted at constant volume, would produce a temperature of 2450° C., absolute.

If the combustion takes place under constant pressure the temperature would be about 2000° absolute. With acetylene this limit may be extended. With other gases slightly different results are obtained, as shown hereafter.

The upper limit of pressure is determined almost wholly by practical considerations. As we shall see, it is without direct influence on the velocity of discharge. If the compression is isothermic it may be increased without increasing the ratio of the work of compression to the useful effect. We may consider compressions of 40 to 60 atmospheres (600 to 900 pounds per square inch) as entirely admissible, both with respect to the compressor and the combustion chamber.

The lower limit of pressure will be that of the atmosphere if the exhaust is made into the open air, or it may be a very low pressure, approaching a perfect vacuum if the discharge is made into a space provided with an air pump.

The ratio of the work of compression to the useful effect $\frac{\mathfrak{T}c}{\mathfrak{T}u}$ plays an important part in the gas turbine, because the compressor, being necessarily distinct from the turbine, its mechanical efficiency η_c (which includes that of the transmission mechanism by which it is driven) has a very marked influence upon the efficiency of the entire machine. When this ratio is very high the importance and bulk of the compressor occasions some practical inconveniences. A ratio approaching unity is practically prohibitory.

Finally, the quantity of useful work produced per kilogramme of gas gives a measure of the specific power of the machine.

These are the principal elements which form the criterion in the discussion which follows. But, before commencing this discussion it remains for us to indicate the hypothesis and the numerical data upon which it is based. In this connection it may be noted that all the computations have been made with the slide rule, this giving a degree of precision quite within the limits of error of the premises.

We begin with the simple and well-known laws of thermodynamics as used by M. Witz in his classical labors upon the gas engine. As a first approximation we assume the specific heat as constant, the value for hot air at constant pressure C_p being taken at the usual value 0.2375, and the specific heat at constant volume c_v being taken as 0.1686, so that their ratio is

$$\gamma=\frac{C_p}{c_v}=1.41.$$

The specific constant of air is taken at 29.3. We neglect the contraction, which may attain 5 per cent.

For the vapor of water we adopt the value $C_p=0.48$. It is only in special cases that we shall take into account the variation of specific heat with the temperature, using the linear formula of M. Lechatelier, $C=a+bT$.

We shall retain for adiabatic expansion the formula of Laplace or Poisson: $pv^\gamma=\text{constant}$.

The modern exponential formulas are very interesting, but their use would burden this discussion to such an extent as to render comparisons impossible.

After having cleared away the general discussion we shall return to the special modifications to which our results must be submitted if we desire to follow the laws of gases to a higher degree of precision.

The Mechanical Efficiency of the Gas Turbine.

We have to consider two distinct machines—the turbine and the compressor, each having its own efficiency. More or less of the heat energy which is transformed into work in the turbine, with the particular efficiency of this machine, is expended in driving the compressor, and the available power is only the difference between the two values.

Thus, let Q be the quantity of heat furnished by the combustion of a kilogramme of gas, q the heat rejected with the exhaust, and ρ the total thermal efficiency, equal by definition, to $\frac{Q-q}{Q}$. If all the losses are reduced to losses of a thermal order there will be produced in work:

$$\mathcal{T}u=\rho EQ$$

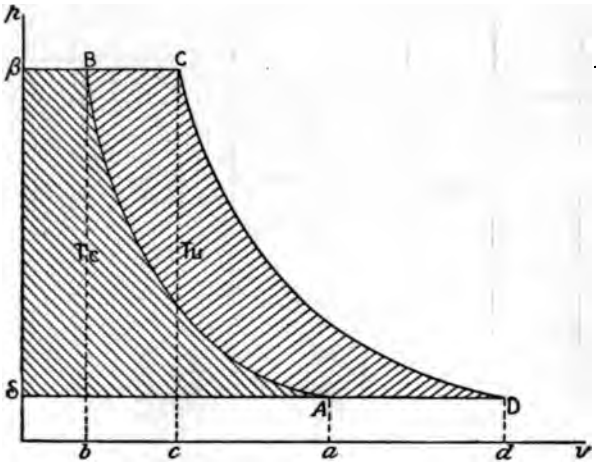


FIG. 39.—Useful work and work of compression.

Now let $\mathcal{T}c$ be the theoretical work of compression per kilogramme of air (this being computed hereafter for each case), and let η_c be the mechanical efficiency of the compressor, defined in such a manner that $\frac{\mathcal{T}c}{\eta_c}$ is the quantity of work delivered to the shaft of the compressor to compress one kilogramme of air.*

On the other hand each kilogramme of air produces in the motor turbine a quantity of "indicated" work, equal, by definition, to the sum of the net available work on the shaft and all the passive resistances (friction, etc.). This

* In the case of isothermal compression the theoretical amount of work may be computed by the law of Mariotte. If this law is not exactly followed and if the gas is slightly heated by the compression the corresponding energy is included in the mechanical losses and in the value of η_c.

work $\mathcal{T}$, which we shall compute for each case, is equal to the useful work $\mathcal{T}u$, defined above, increased by the work of compression $\mathcal{T}c$, as we shall see by examining the diagram, Fig. 39.

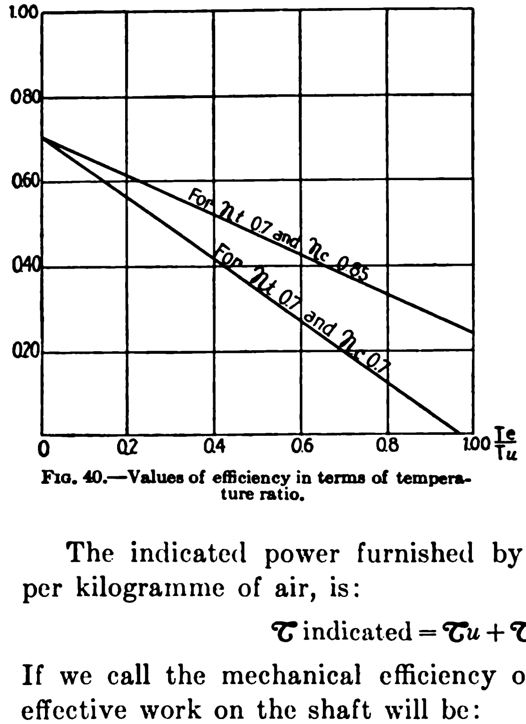


Fig. 40.—Values of efficiency in terms of temperature ratio.

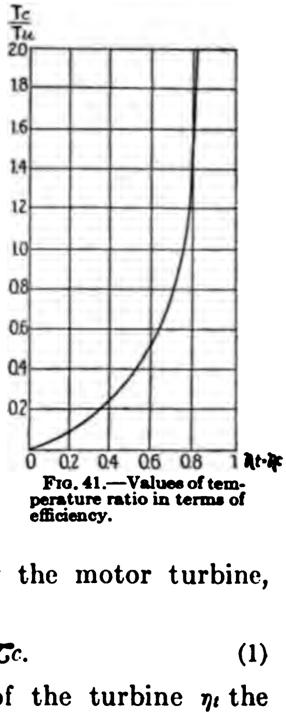


Fig. 41.—Values of temperature ratio in terms of efficiency.

The indicated power furnished by the motor turbine, per kilogramme of air, is:

$$\mathcal{T}\text{ indicated} = \mathcal{T}u + \mathcal{T}c. \tag{1}$$

If we call the mechanical efficiency of the turbine η_t the effective work on the shaft will be:

$$\mathcal{T}\text{ effective} = \eta_t(\mathcal{T}u + \mathcal{T}c). \tag{2}$$

It follows that the effective work available upon the common shaft of the turbine and the compressor, supposing them to be connected thus as one machine, will be:

$$\mathcal{T}\text{ net work} = \eta_t(\mathcal{T}u + \mathcal{T}c) - \frac{1}{\eta_c}\mathcal{T}c. \tag{3}$$

If the compressor be driven through any intermediate transmission the efficiency of this transmission should be included in η_c.

The mechanical efficiency of the two machines together will then be:

$$\eta = \frac{\text{net work}}{Tu} = \eta_t + \left(\eta_t - \frac{1}{\eta_c}\right)\frac{\mathfrak{T}c}{\mathfrak{T}u} \tag{4}$$

The mechanical efficiency then disappears when

$$\frac{\mathfrak{T}c}{\mathfrak{T}u} = \frac{\eta_t}{\dfrac{1}{\eta_c} - \eta_t}$$

For example, taking $\eta_t = \eta_c$ the efficiency will be zero for

$$\frac{\mathfrak{T}c}{\mathfrak{T}u} = \frac{\eta_t^2}{1 - \eta_t^2}$$

which gives the following values:

$\eta_t = \eta_c =$	0.5	0.6	0.7	0.8
$\dfrac{\mathfrak{T}c}{\mathfrak{T}u} =$	0.34	0.56	0.96	1.78

We shall see later on that under the actual conditions of turbine construction η_t will have a value of about 0.7.

As for the efficiency of the compressor η_c, this will be for improved reciprocating machines 0.8 to 0.9.

Since it is necessary, however, to introduce a speed-reduction transmission, η_c will be reduced to 0.75 to 0.85.

If the compressor is made of the multicellular turbine type, permitting direct connection, it is possible that the efficiency will be in the neighborhood of 0.6 to 0.7.

If we take $\eta_t = \eta_c = 0.7$ we see that the mechanical efficiency will be totally annulled for

$$\mathfrak{T}c = \mathfrak{T}u, \text{ about.}$$

We have in general

$$\eta = 0.700 - 0.729\frac{\mathfrak{T}c}{\mathfrak{T}u}$$

This shows the fundamental importance of the ratio of the work of compression to the useful work.

In order to establish our ideas in this respect, we may consider theoretically that this ratio will lie somewhere between 0.2 and 0.4, which will cause the total mechanical efficiency to range between 0.4 and 0.6. As we shall find the thermal efficiency to lie between 0.4 and 0.6 we see that the total useful efficiency will be from 0.16 to 0.36.

We shall now pass to the discussion of the various cycles applicable to the gas turbine.

I.

A. Cycles Using the Isothermic Introduction of Heat.

The typical cycle of this group is that of Carnot. Diesel has sought to use this by realizing isothermal combustion in his motor. This result can be obtained in a gas turbine only by causing the combustion to be continued in the expansion nozzle, or by causing the expansion to take place in several stages with successive interheaters. This last solution, however, would only be an approximative one.

The Carnot Cycle.

The kilogramme of gas under consideration is compressed from p_0 to p_1 maintaining at the same time the initial temperature T_0. This isothermal compression absorbs a quantity of work $\mathfrak{T}_i$, given by the equation:

$$\mathfrak{T}_i = RT_0 \log \text{hyp}\left(\frac{p_1}{p_0}\right).$$

The gas is now compressed adiabatically from p_1 to p_2. The temperature passes from T_0 to T_2 and the work absorbed by the compression is

$$\mathfrak{T}_a = EC_p(T_2 - T_0).$$

We also have:

$$\frac{p_2}{p_1} = \left(\frac{T_2}{T_0}\right)^{\frac{\gamma}{\gamma-1}}.$$

We then introduce the quantity of heat Q upon the isothermal CD, at the temperature T_2, during which period the pressure falls from p_2 to p_3.

We then have:

$$Q = ART_2 \log\left(\frac{p_2}{p_3}\right).$$

We know that the thermal efficiency of the Carnot cycle is equal to:

$$1 - \frac{T_0}{T_2}$$

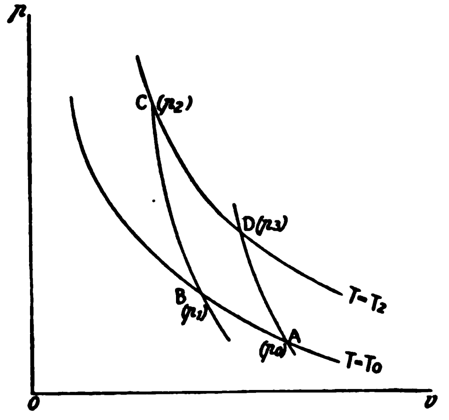


FIG. 42.—The Carnot cycle.

and that the useful work is:

$$\mathfrak{T}u = EQ\left(1 - \frac{T_0}{T_2}\right).$$

We also have:

$$\frac{p_1}{p_0} = \frac{p_2}{p_3} = e^{\frac{Q}{ART_2}}$$

The properties of the cycle depend only upon the temperature of combustion T_2, the total compression ratio $\frac{p_2}{p_0}$, and the introduction of the heat Q. The temperature of the exhaust is that of the atmosphere, about 300° C. absolute.

The thermal efficiency, which depends only upon T_2, may reach very high theoretical values, but to attain these involves the use of excessively high compressions; thus:

Temperature of combustion T_2	300	600	900	1200	1500	1800	2100
Thermal efficiency ρ	0	0.50	0.66	0.75	0.80	0.83	0.86
Adiabatic compression ratio $\frac{p_2}{p_1}$	1	11	46	128	282	525	913

We see that it is impossible to pass a thermal efficiency of 0.66 without being obliged to have recourse to excessive compressions, since it is necessary to multiply the adiabatic compression ratio, which we shall calculate.

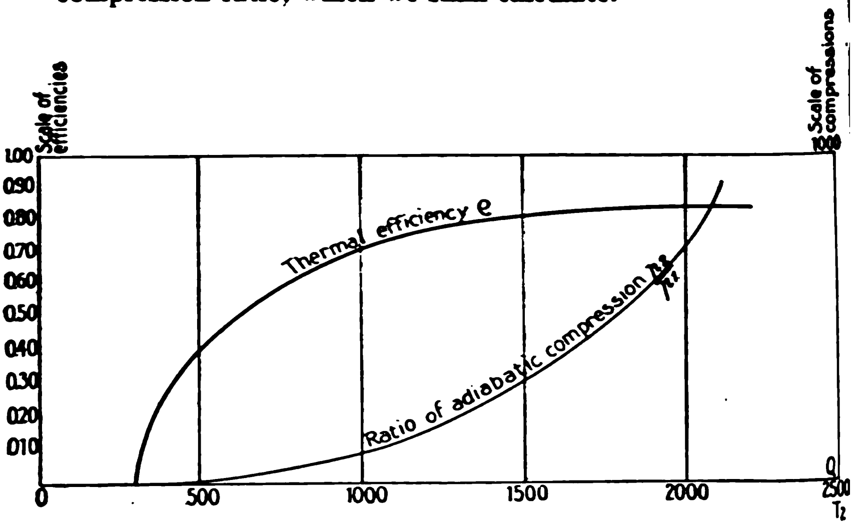


FIG. 43.—Efficiency and ratio of adiabatic compression. Carnot cycle.

This latter: $\frac{p_0}{p_1}$ is a function of $\frac{Q}{T_2}$. For $T_2 = 900$ degrees, and $Q = 300$ calories, we have: $\frac{Q}{T_2} = 0.333$, and

$$\frac{p_1}{p_0} = 120.$$

The ratio of the work of compression to the useful work is given by:

$$\frac{\mathfrak{T}c}{\mathfrak{T}u}=\frac{\mathfrak{T}i}{\mathfrak{T}u}+\frac{\mathfrak{T}a}{\mathfrak{T}u}=\frac{RT_0 \log\left(\frac{p_1}{p_0}\right)+EC_p(T_2-T_0)}{\mathfrak{T}u}$$

$$=\frac{RT_0\frac{Q}{ART_2}+EC_p(T_2-T_0)}{EQ\left(1-\frac{T_0}{T_2}\right)}$$

whence:

$$\frac{\mathfrak{T}c}{\mathfrak{T}u}=\frac{1}{\frac{T_2}{T_0}-1}+C_p\frac{T_2}{Q}.$$

In our particular case we have:

$$\frac{\mathfrak{T}c}{\mathfrak{T}u}=1.2.$$

We have seen above that the mechanical efficiency disappears as the ratio between the work of compression and the useful work approaches unity. As a matter of fact we cannot even admit sufficiently high values for the thermal efficiency and for the temperature T_2, since the latter, resulting from the adiabatic compression, cannot exceed 700 degrees C. absolute, whether we employ a reciprocating piston compressor or a rotary turbine compressor.

The Carnot cycle is therefore not adapted to the gas turbine, since the high thermal efficiency which can be realized by its use is obtainable only by the employment of very high compressions and enormous masses of gas. In consequence, the compressor, which is admittedly the weak point of the gas turbine, assumes an excessive importance, and the mechanical losses would absorb all the useful work.

The Diesel Cycle.

Theoretically the cycle of Diesel differs from the Carnot cycle by the substitution of a wholly adiabatic compression for the two successive compressions, isothermal and adiabatic, of Carnot. The rejection of heat to the cooling medium is thus produced by the non-closure of the cycle:

Here again the isothermal expansion is defined by:

$$\frac{p_2}{p_3} = e^{\frac{Q}{ART_1}}$$

and a considerable degree of expansion is required to enable a sufficient quantity of heat Q, to be introduced.

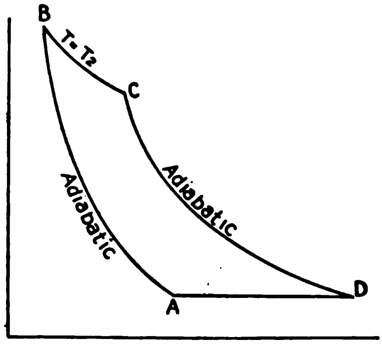


FIG. 44.—The Diesel cycle.

Even if we admit that the temperature of combustion, obtained at the end of the adiabatic compression, may attain 800 degrees (corresponding to a compression of 35 atmospheres), we have, for:

$Q =$	100	200	300 calories
$\frac{p_2}{p_3} =$	6	37	220

We cannot therefore exceed an introduction of 200 calories, and even at this figure there would no longer be an adiabatic expansion.

The maximum temperature of the cycle should therefore occur at the beginning of the combustion, and should be superior to that produced by the compression, and thus the curve of combustion should keep above the isothermal.

In any case this cycle is not adapted to the gas turbine.

Partial Isothermal Cycles.—Some writers, Barkow among others, have suggested that the combustion should be started under constant pressure, and completed isothermically. We shall examine this solution later on, but it is difficult of realization in turbines, and offers no especial advantages.

B. Cycles Using the Isobaric Introduction of Heat.

Combustion under Constant Pressure.—With combustion at constant pressure the temperature of the gas is raised. It is preceded by a compression which may be either adiabatic or isothermic. In the first case the compression is not accompanied by the transfer of any heat to the cooling medium, but it involves the expenditure of a greater amount of work. A complete computation is necessary to show which of the two systems should be adopted. The following table will serve as a basis for the calculations:

Compression ratio.	5	10	15	20	25	30	40	60	80	100
Final temperature of adiabatic compression ..	479	585	658	716	764	804	875	990	1040	1150
Equivalent in calories of work of compression: adiabatic	42	68	85	99	110	120	136	164	176	203
Equivalent in calories of work of compression: isothermal ..	33	48	56	62	67	71	76	85	90	95
Ratio of the two efforts........................	0.78	0 70	0.66	0.62	0.61	0.59	0.56	0.52	0.51	0.48

These figures are calculated upon the assumption of an initial temperature of 300 degrees C. absolute, and result in the following considerations:

The work absorbed by the compressor consists of the compression, properly so-called, which is given, in the case when operating at constant temperature, by the formula

$$RT_0 \log\left(\frac{p_1}{p_0}\right).$$

and the work necessary to drive the compressed air into the reservoir at the pressure p_1; that is $p_1\, v_1 - p_0\, v_0$.

But in this case the second term is zero, and we have:

$$\mathcal{T}i = RT_0 \log\left(\frac{p_1}{p_0}\right).$$

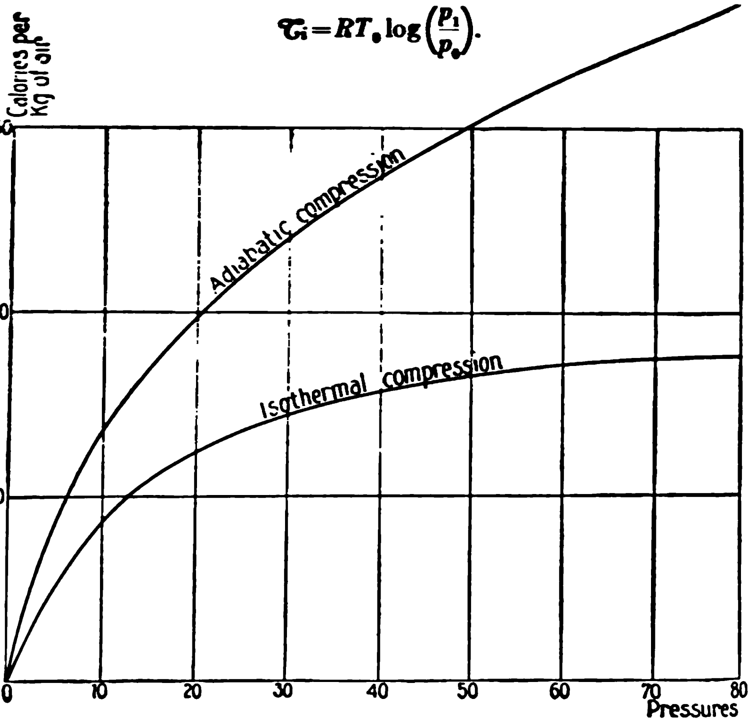


FIG. 45.—Equivalent in calories of the work of compression per kilogramme of air.

In the case of the adiabatic compression the first term has a value $EC_v(T_1-T_0)$, and the second $(p_1v_1-p_0v_0)$ is equal to $R(T_1-T_0)$, whence:

$$\mathcal{T}a = Ec_v + R(T_1 - T_0)$$

and since $$R = E(C_p - c_v)$$

we have: $$\mathcal{T}a = EC_p(T_1 - T_0).$$

From this it follows that:

$$\frac{\mathcal{T}i}{\mathcal{T}a} = \frac{R}{EC_p} \frac{\log\left(\frac{p_1}{p_0}\right)}{\left(\frac{p_1}{p_0}\right)^{\frac{\gamma-1}{\gamma}} - 1} = \frac{\gamma - 1}{\gamma} \frac{\log\left(\frac{p_1}{p_0}\right)}{\left(\frac{p_1}{p_0}\right)^{\frac{\gamma-1}{\gamma}} - 1}.$$

This ratio tends to approach unity for infinitely small compressions. It decreases rapidly as the compression ratio increases, and falls to 0.5 for a compression of about 80.

The power absorbed by the compressor is therefore less, for the same compression, with the isothermal method than with the adiabatic. This difference is still more marked if, for any reason, the gas which has been compressed adiabatically is allowed to return to its initial temperature. Thus a compression of 20 will drop to about 8.4. This fact renders adiabatic compression inadmissible for the ordinary applications of compressed air. In the case of the gas turbine, however, the sensible heat of the adiabatically compressed gas is not lost, and a fuller discussion of the subject becomes necessary.

Adiabatic Compression.

During the compression the pressure passes from p_0 to p_1 and the temperature from T_0 to T_1.

We have:
$$\frac{p_1}{p_0}=\left(\frac{T_1}{T_0}\right)^{\frac{\gamma}{\gamma-1}}.$$

The introduction of Q calories, at the constant pressure p_1, raises the temperature to T_2, the temperature of combustion, and

$$Q=C_p\,(T_2-T_1).$$

The mixture then expands adiabatically from T_2 to T_3, and

$$\frac{T_3}{T_2}=\left(\frac{p_0}{p_1}\right)^{\frac{\gamma-1}{\gamma}}.$$

We see that:

$$\frac{T_1}{T_0}=\frac{T_2}{T_3}.$$

The quantity of heat rejected to the cooling medium is equal to the heat carried off by the exhaust gases, that is:

$$q=C_p(T_3-T_0).$$

The thermal efficiency ρ will then be:

$$\rho=\frac{Q-q}{Q}=1-\frac{T_0}{T_1}.$$

We therefore obtain the same efficiency as in a Carnot cycle having the same ratio of adiabatic compression, but without having the necessity for the preliminary isothermal compression.

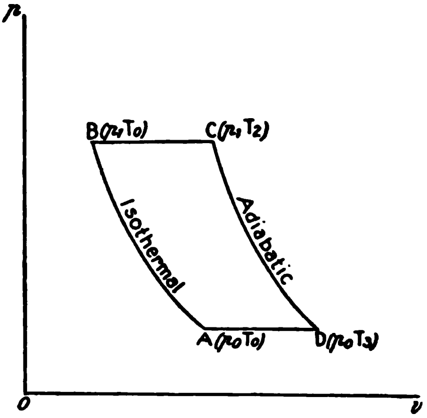


FIG. 46.—Cycle of isobaric combustion with adiabatic compression.

The total compression is therefore much lower, but the upper temperature of the cycle is much higher, a matter which offers no inconvenience.

The ratio of the work of compression to the useful work is given by

$$\frac{\mathfrak{T}c}{\mathfrak{T}u}=\frac{EC_p(T_1-T_0)}{EQ\left(1-\frac{T_0}{T_1}\right)}=C_p\frac{T_1}{Q}.$$

It is easy to see that this ratio is constant if we give the temperature T_3 at the end of the expansion a fixed value, for we have $\frac{T_2}{T_3}=\frac{T_1}{T_0}$, and consequently:

$$\frac{\mathfrak{T}c}{\mathfrak{T}u}=\frac{C_pT_1}{C_p\left(\frac{T_3T_1}{T}-T_1\right)}=\frac{1}{\frac{T_3}{T_0}-1}.$$

This ratio attains greater value, therefore, as the temperature T_3 has a higher value. Since, however, for constructive reasons, T_3 cannot be allowed to exceed 700 degrees C., we have:

$$\frac{\mathfrak{T}c}{\mathfrak{T}u}=\frac{1}{\frac{700}{300}-1}=0.75$$

The corresponding value of the total mechanical efficiency η, given by the equation $\eta=0.700-0.729\frac{\mathfrak{T}c}{\mathfrak{T}u}$, is therefore only about 0.15.

The properties of the most advantageous family of cycles are given below:

Ratio of compression $\frac{p_1}{p_0}$	5	10	15	20	30
Final temperature of compression T_1	480	585	658	716	804
Final temperature of combustion T_2	1120	1400	1540	1670	1876
Final temperature of expansion T_3	700	700	700	700	700
Heat introduced (calories) Q	149	188	205	227	255
Heat lost in the exhaust q	92	92	92	92	92
Thermal efficiency ρ	0.37	0.49	0.54	0.58	0.63
Equivalent $A\mathfrak{T}c$ of work of compression	42	68	85	99	120
Equivalent $A\mathfrak{T}u$ of useful work	56	94	112	134	162
Total useful effect $\rho\eta$	0.06	0.08	0.086	0.092	0.10
Equivalent $A\eta\mathfrak{T}u$ of net mech. work	8.4	14.1	16.8	20	24.3
Consumption of air per H.P. hour, kg.	75	45	37	32	26
Ratio of powers $\frac{Nc}{Nu}$	7.1	7.1	7.1	7.1	7.1

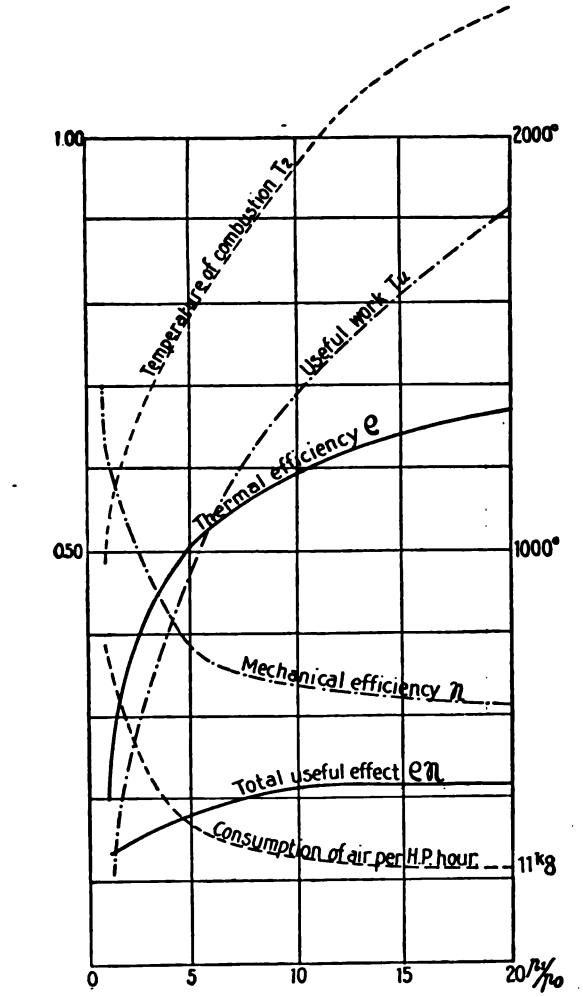
These results, plotted in the diagram, are not very encouraging. An adiabatic compression of 20 gives a final temperature of 716, which should not be exceeded.*

The useful effect does not exceed 9 per cent., and the mass of gas required to produce a unit of work is considerable.

* That is, if the action is truly adiabatic, without any artificial cooling of the parts in contact with the gas. If this is not the case it is impossible to calculate accurately the results which may be attained.

Cycles with Isobaric Combustion and Isothermal Compression.

Compression ratio $\frac{p_1}{p_2}$	5	10	15	20	25	30	40	60	80	100
Temperature of combustion T_2	1120	1365	1533	1680	1780	1880	2050	2300	2500	2670
Heat introduced (calories) Q	195	254	292	328	350	375	415	480	522	563
Heat lost in the exhaust q	92	92	92	92	92	92	92	92	92	92
Heat lost in the compression Q'	33	48	56	62	67	71	76	85	90	95
Thermal efficiency ρ	0.36	0.45	0.49	0.53	0.545	0.565	0.595	0.63	0.65	0.67
Equivalent $A\mathfrak{T}c$ of work of compression	33	48	56	62	67	71	76	85	90	95
Equivalent $A\mathfrak{T}u$ of useful work	70	114	144	174	191	212	247	303	339	377
Ratio $\frac{\mathfrak{T}c}{\mathfrak{T}u}$	0.47	0.42	0.39	0.36	0.35	0.34	0.31	0.28	0.26	0.25
Mechanical efficiency η	0.36	0.39	0.42	0.44	0.445	0.45	0.47	0.49	0.51	0.52
Total useful effect $\rho\eta$	0.13	0.18	0.205	0.233	0.243	0.255	0.28	0.31	0.33	0.35
Ratio of powers $\frac{Nc}{Nu}$	1.8	1.6	1.3	1.2	1.1	1.06	0.95	0.83	0.72	0.68



-Cycles with adiabatic compression and isobaric combustion. Exhaust discharged at 700 degrees C.

Isothermal Compression.

e isothermal compression from p_0 to p_1, effected at a rature T_0, absorbs a quantity of work of which the lent in heat is:

$$Q' = ART_0 \log \text{hyp}\left(\frac{p_1}{p_0}\right).$$

The introduction of Q calories, at the constant pres
p_1, raises the temperature to T_2, and:

$$Q=C_p(T_2-T_0).$$

The adiabatic expansion from p_1 to p_0 brings the temp
ture to T_3, and

$$\frac{T_3}{T_2}=\left(\frac{p_0}{p_1}\right)^{\frac{\gamma-1}{\gamma}}.$$

The exhaust gases carry off with them heat equa
calories to:

$$q=C_p(T_3-T_0).$$

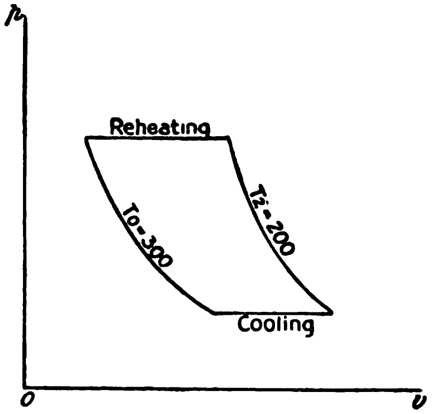


FIG. 48.—Cycle with isobaric combustion and isothermal compression.

The thermal efficiency will therefore be:

$$\rho=\frac{C_p(T_2-T_0)-C_p(T_3-T_0)ART_0\log\text{hyp}\left(\frac{p_1}{p_0}\right)}{C_p(T_2-T_0)}.$$

The table on page 125 gives the results of computat
for a series of cycles having the same temperature of exh
$T_3=700°$ C.

These results show that it is important to use as hi
compression as possible. In this respect the limits
those imposed by the values for the temperatures of (

bustion and of the introduction of the heat, these becoming excessive when the total compression passes 60.

In this case the results may be ameliorated by admitting a lower temperature for the exhaust. For example, let us take a compression of 80. Let the temperature of the exhaust be placed at 600° absolute, instead of 700°.

The temperature of combustion will then be 2150°. We also find that the temperature Q of introduction of the heat will be 440, the efficiency 0.635, the ratio $\frac{\mathfrak{T}c}{\mathfrak{T}u}=\frac{90}{279}=$ 0.325, whence $\eta=0.46$, and consequently $\rho\eta=0.29$; while we should have had 0.28 with a compression of 40, and an exhaust temperature of 700° absolute.

There would be no theoretical advantage in thus lowering the temperature of the exhaust, if it did not have the practical value of aiding in the use of successive pressure stages in the turbine. By the use of this system, however, we may carry the expansion down to 700° in the nozzles of the first turbine, and continue it from 700° to 600° in the guide blades of a second group of revolving wheels.

A series of practical tests would be required to determine the exact limit from which high compressions could be utilized in this manner. This limit will be found to depend both upon the true law governing the expansion and upon the temperature of combustion actually attainable*.

Below this limit it will not be found advantageous to attempt to utilize high compressions by the reduction of the temperature of the exhaust, and it will be a better endeavor to increase the introduction of the heat and the maximum temperature.

It does not seem to be a practical impossibility to realize higher compression ratios than 50. The construction of the compressors and the combustion chambers for such pressures seems practicable.

* The values for γ, and for the specific heats, may differ in practice from those which have been taken in the computations.

But the actual pressure may be modified by causing the exhaust gases to be discharged at a lower pressure. This point will be considered more fully hereafter, and it is now only necessary to mention that the lower pressure p_0 may be made less than atmospheric pressure (such as $\frac{1}{3}$, $\frac{1}{4}$, or $\frac{1}{5}$ atmosphere) without in any way changing the preceding arguments and calculations. The upper pressure limit p_1 will then be reduced to $\frac{1}{3}$, $\frac{1}{4}$, or $\frac{1}{5}$ of its former value. The ratio $\frac{p_1}{p_0}$ will not be changed, and the efficiencies will not be modified.

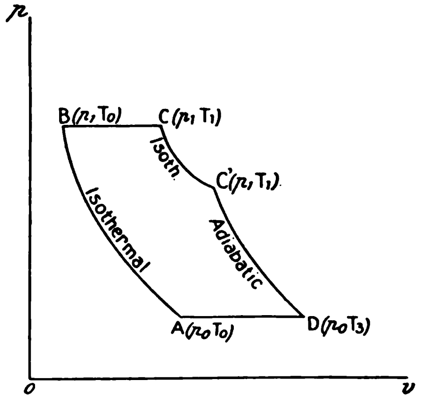


FIG. 49.—Cycle with isobaric combustion and isothermal expansion.

We may mention here a modification suggested by Barkow among others, in which the combustion, commenced under constant pressure, is completed in the course of an isothermal expansion.

Let us suppose the adiabatic expansion is the same as in the preceding case, and the final temperature 700° C. absolute. It follows that the upper temperature limit will be the same, and consequently also the quantity of heat Q, introduced under constant pressure. But we may introduce a supplementary quantity of heat K, during the isothermal expansion. Let χ be the ratio of this expansion, we will then have: $K = ART_2 \log \text{hyp} (\chi)$.

The total compression will be χ-times greater than before, so that the work of compression will be increased by a supplementary amount

$$\mathcal{T}'_c = ART_0 \log \text{hyp} (\chi).$$

There will then be a gain of work equal to:

$$AR(T_2 - T_0) \log \text{hyp} (\chi) \text{ or } \frac{T_2 - T_0}{T_2} K.$$

The quantity of heat introduced at constant temperature is then utilized with a thermal efficiency equal to that of the Carnot cycle, so that the efficiency of the entire cycle is improved. It is necessary, however, to give a considerable value to χ, in order that K may obtain any importance. A complete computation shows that it would be better, so far as the total useful effect is concerned, to utilize all the compression available to raise to a maximum the introduction of heat under constant pressure.

Discussion of Comparative Efficiencies.

We may now make a definite comparison of the two modes of compression.

It will be seen at once, by an inspection of the diagram, that the thermal efficiency ρ is slightly greater when we use adiabatic compression. But since, with this system we cannot exceed in practice a compression ratio of 20, the maximum value for ρ is 0.58; while when the compression is isothermal, we may carry the compression as high as 60, which gives for ρ the maximum value 0.63.

The superiority of isothermal compression is still more marked from the point of view of the mechanical efficiency. While this remains constant whatever the compression in the adiabatic system, it increases with the compression in the isothermal system, and attains values ranging from double to triple those realized in the first case. The same is practically true with regard to the total efficiency $\rho\eta$, which, according to our hypotheses, appears to have a limit of about 0.30.

It will, therefore, be necessary to expend 2120 calories per effective horse-power hour, delivered on the shaft, which corresponds to a consumption of 212 grammes of hydro-carbon fuel, having a calorific value of 10,000 calories (lower calorific value). The Diesel and the Banki motors have a consumption of 180 to 250 grammes. Gas engines operating with blast-furnace gas require at a minimum about 2000 calories per effective horse-power.

The fuel consumption of the gas turbine is therefore comparable with that of the best motors known. The weak point appears in the fact that the effective power absorbed by the compressor is equal to about 85 per cent. of the net effective power practically available on the shaft.

It should be noted that if, by reason of the defective arrangement of the compressor, the heat developed during the compression is not immediately absorbed by the injection of water and by cooling the walls, but only disappears in the flow of the gas between the compressor and the turbine, the efficiency will be much less than in the two preceding cases.

We have, in fact, the following results: the ratio of the work of compression to the useful work will be constant whatever the degree of compression for any given temperature of exhaust. For an exhaust temperature of 700° C. absolute, this ratio is 0.75, whence $\eta = 0.17$. As for the thermal efficiency, it will vary as shown in the following table:

Ratio of compression.	5	10	15	20	30	40	60	80	100
Heat introduced, Q	195	254	292	328	375	415	480	522	563
Heat lost in compression, Q'	42	68	85	90	120	136	164	176	202
Equivalent $A\mathfrak{T}u$, useful work	61	94	115	137	163	187	224	254	271
Thermal efficiency ρ........	0.31	0.37	0.39	0.44	0.44	0.45	0.47	0.49	0.49
Total useful effect $\rho\eta$	0.053	0.063	0.067	0.071	0.074	0.077	0.080	0.083	0.080

The results would not be as low as indicated in the table if the compression were effected in several stages, because the cooling of the gas between the cylinders reduces the expenditure of work. It might be possible to obtain a satisfactory result if the compression were divided into a number of stages, with complete inter-cooling.

Cycles with Isopleric Introduction of Heat.

(Cycles for Explosion Motors.)

Without discussing, for the moment, whether or not this method is practically applicable to the gas turbine we may examine the efficiencies which it is theoretically capable of realizing.

The introduction of heat at a constant volume causes an increase of pressure, and produces a greater rise of temperature than if a constant-pressure system is employed.

Adiabatic Compression.

If the compression is effected adiabatically the final temperature of the explosion will be very high, and the introduction of heat per kilogramme of gas cannot be very great. In fact we are obliged to require excessive final

Ratio of compression $\frac{p_1}{p_0}$	1	5	10	15	20
Final temperature compression T_1	300	480	585	655	717
Final temperature combustion T_2	980	1530	1930	2190	2300
Heat introduced Q (calories)	115	188	230	260	270
Thermal efficiency ρ	0.200	0.505	0.595	0.645	0.654
Equivalent $A\mathfrak{T}c$ of work of compression	0	42	68	85	99
Equivalent $A\mathfrak{T}u$ of useful work	23	96	138	168	178
Ratio $\frac{\mathfrak{T}c}{\mathfrak{T}u}$	0	0.44	0.50	0.52	0.55
Mechanical efficiency η	0.70	0.38	0.34	0.32	0.30
Total useful effect $\rho\eta$	0.140	0.192	0.203	0.210	0.200
Ratio of power $\frac{Nc}{Nu}$	0	1.66	2.1	2.3	2.6
Ratio of pressures $\frac{p_2}{p_0}$	3.27	15.4	32.7	49	65
Net available mechanical work $\eta\mathfrak{T}u$	16	36.5	47.0	54.0	54.0
Consumption of air, kg. per H.P. hour	39.5	17.4	13.6	11.8	11.8

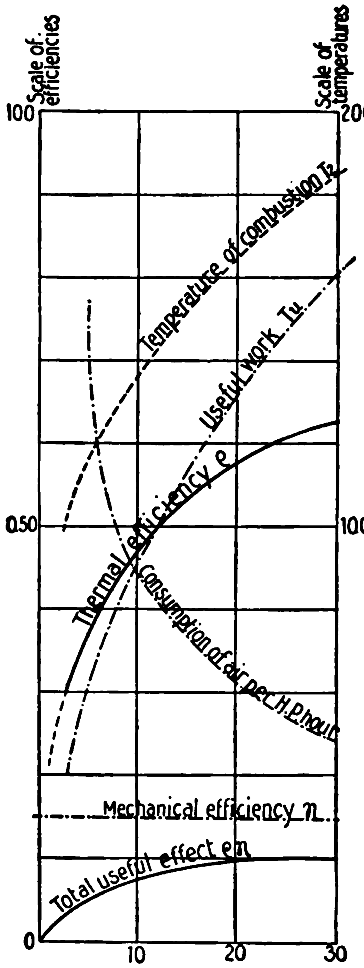


FIG. 50.—Cycle with adiabatic compression and isopleric combustion. Exhaust esca at 700 degrees C. absolute.

explosion pressures in order to attain the temperature 700° at the end of the expansion. It is therefore impracti ble to exceed a compression ratio of 15, which leads to explosion pressure of 49 atmospheres. Under these con tions, themselves difficult to realize, the thermal efficier

ρ will be about 0.64, with a mechanical efficiency of 0.33 and a useful effect of 0.21, as shown in the table on page 133, which gives, as before, the results of a series of cycles, for an exhaust temperature of 700° C. absolute.

Explosion turbines with adiabatic compression have, therefore, a low efficiency, the total useful effect not exceeding 20 per cent. Increase of initial compression has but a slight influence, so that such machines are of interest only for small powers, or in cases in which the consumption of fuel is a secondary consideration.

Isothermal Compression.

In this case we introduce a quantity of heat Q for each kilogramme of gas and have:

$$Q = c_v(T_2 - T_0).$$

The pressure becomes

$$\frac{p_2}{p_1} = \frac{T_2}{T_0}.$$

The adiabatic expansion brings the temperature down to T_3.

We then have:

$$\frac{T_3}{T_2} = \left(\frac{p_0}{p_2}\right)^{\frac{\gamma-1}{\gamma}}$$

and

$$T_2 = \frac{(T_3)^{\gamma}}{\left(\frac{p_0}{p_1}T_0\right)^{\gamma-1}}.$$

The heat discharged to the cooling medium is composed of the calories rejected with the exhaust: $C_p(T_3 - T_0)$ and the heat subtracted during the isothermal compression: $RT_0 \log \operatorname{hyp}\left(\frac{p_1}{p_0}\right)$. The thermal efficiency will then be:

$$\rho = \frac{c_v(T_2 - T_0) - C_p(T_3 - T_0) - RT_0 \log \operatorname{hyp}\left(\frac{p_1}{p_0}\right)}{c_v(T_2 - T_0)}.$$

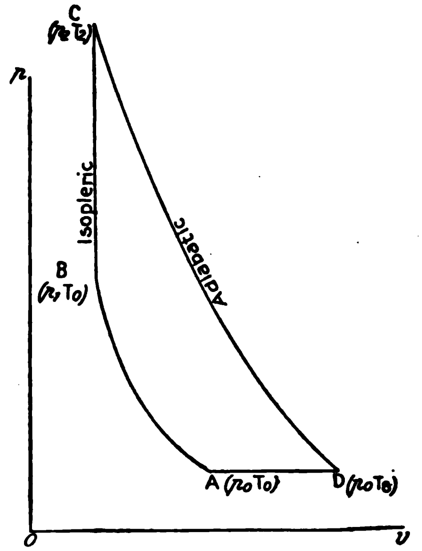


FIG. 51.—Cycle with isopleric combustion and isothermal compression.

If, as before, we take the exhaust temperature at 700° C. abs., we have the corresponding family of cycles as follows:

Compression ratio $\frac{p_1}{p_0}$	1	5	10	15	20
Temperature of compression T_2	980	1820	2420	2850	3210
Heat introduced Q (calories)	115	255	355	430	490
Heat lost in the exhaust q	92	92	92	92	92
Heat lost in the compression	0	33	48	56	62
Thermal efficiency ρ	0.19	0.51	0.61	0.66	0.68
Equivalent $A\mathcal{T}c$ of work of compression	0	33	48	56	62
Equivalent $A\mathcal{T}u$ of useful work	23	130	215	282	336
Ratio $\frac{\mathcal{T}c}{\mathcal{T}u}$	0	0.25	0.22	0.20	0.19
Mechanical efficiency η	0.70	0.52	0.54	0.55	0.56
Total useful effect $\rho\eta$	0.13	0.265	0.33	0.365	0.38
Ratio of powers $\frac{Nc}{Nu}$	0	0.68	0.58	0.52	0.49
Ratio $\frac{p_2}{p_1}$	3.27	6.1	8.1	9.5	10.7
Ratio $\frac{p_2}{p_0}$	3.27	30.4	80.7	143	214
Net available mechanical work $\eta\mathcal{T}u$	16	68	116	156	188
Consumption of air, kg. per H.P. hour	39.5	9.4	5.5	4.1	3.4

The ratio of the work of compression to the useful work is low, which is a great advantage. But with even a compression ratio of 10 the final pressure passes 80 atmospheres, a limit very difficult to handle. The total useful effect then reaches 0.38, while with isothermal compression followed by combustion at constant pressure the limit is 0.31.

If the exhaust is discharged into a space having a reduced pressure it becomes practicable to use higher pressure ratios. Thus, with a compression of 3 and an introduction of 430 calories, the maximum pressure reaches 28.5 atmospheres. If we allow this to escape into a space having a pressure of $\frac{1}{5}$ atmosphere we get a total expansion of 143 times, this being necessary to reduce the temperature of the gas from 2850° to 700° absolute, which gives a useful effect of 0.365, while the power of the compressor will be reduced to about one-half the net available power. These results are very encouraging. Unfortunately, it is not easy to construct a satisfactory explosion turbine, the operative portions of the explosion chamber being unable to resist the very high temperatures developed.

It may readily be shown that the efficiency becomes less favorable if the temperature of the exhaust is made lower than 700 degrees.

Isopleric Combustion Cycles without Compression.

If the gas is not compressed before the explosion the efficiency is low, as we have already seen. We may investigate the manner in which it varies if the temperature T_3 of the exhaust is varied.

We have:

$$\rho = 1 - \gamma \frac{T_3 - T_0}{T_2 - T_0} = 1 - \gamma T_0^{\gamma - 1} \frac{T_3 - T_0}{T_3^{\gamma} - T_0^{\gamma}}.$$

It is easy to see that efficiency will be a minimum when $T_3 = T_0$, and increases with the increase of T_3 above its minimum value T_0.

For	$T_3=400$	600	800	1000
We have	$\rho=0.06$	0.16	0.23	0.27
and hence	$\rho\eta=0.04$	0.11	0.16	0.17.

We see that the highest efficiency corresponds to the highest temperature of the exhaust admissible, with regard to the endurance of the metallic turbine wheel. Under the most favorable conditions the total efficiency cannot be expected to surpass 14 per cent.

II.

Cycles with Expansion Prolonged below Atmospheric Pressure.

In the cycles thus far examined the gas has been expanded down to the pressure of the atmosphere and rejected at a temperature dependent upon the conditions of operation; chosen, however, as high as possible, with respect to

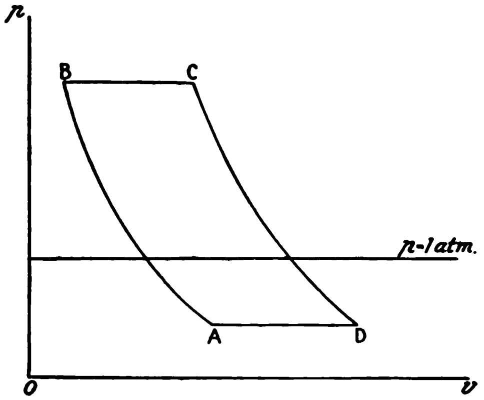


FIG. 52.—Cycle with prolonged expansion.

the endurance of the turbine wheel. There is, however, nothing to prevent the arrangement of the parts in such a manner as to cause a part of the process to be conducted at a pressure below that of the atmosphere; as has already been done in the so-called "atmospheric" gas engines, using a free piston.

In the case of the turbine this may be effected by the use of an air pump. When, however, the large dimensions are considered, a piston pump is seen to be unsuited for this purpose. It is necessary, therefore, to use multicellular turbine machines similar to those already designed for compressors. For a reduction of the pressure to $\frac{1}{5}$ atmosphere it will not be found necessary to use more than five turbine wheels. In order, however, to reduce the amount of work absorbed by this machine it is necessary that it should be

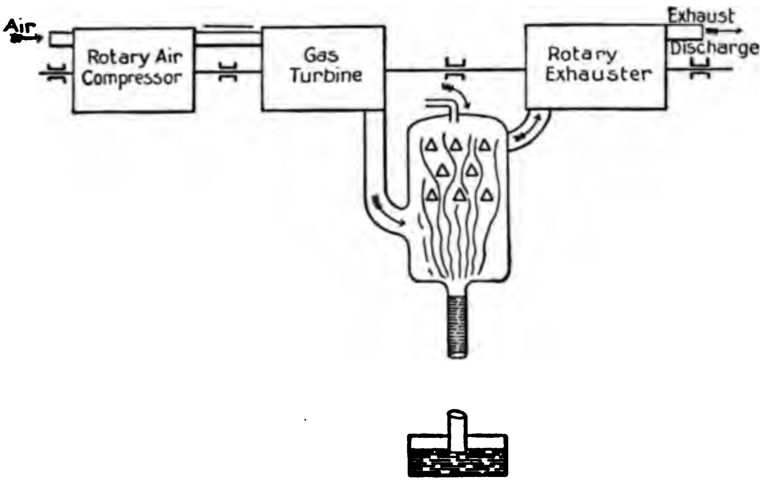


FIG. 53.—Turbine with exhaust under reduced pressure without regenerator.

operated at a constant temperature, and it is desirable that the temperature of the exhaust gases should be brought down to 300° C. absolute, before these gases enter the suction blower, because the work absorbed by the machine, $RT \log (\chi)$, is proportional to the absolute temperature of the gases.

This result may be obtained by cooling the gases by means of an abundant injection of water, in connection with the use of a sort of barometric condenser (Fig. 53), or by the use of a tubular refrigerator with circulating water (Fig. 54).

It is evident that this latter method may be used in connection with some system of regeneration. The gases may also be cooled in the vaporizer of sulphurous-acid gas, forming part of a refrigerating machine, as we shall see hereafter (Fig. 55).

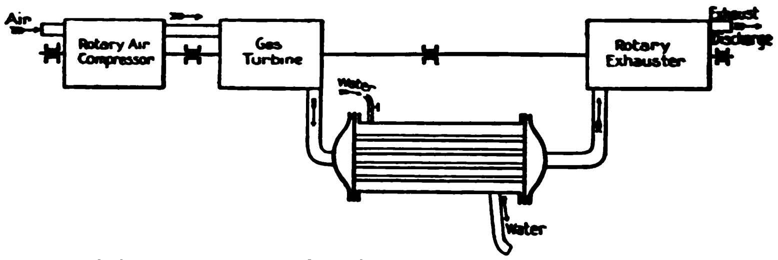


FIG. 54.—Turbine with exhaust under reduced pressure, without regenerator, with intercooler.

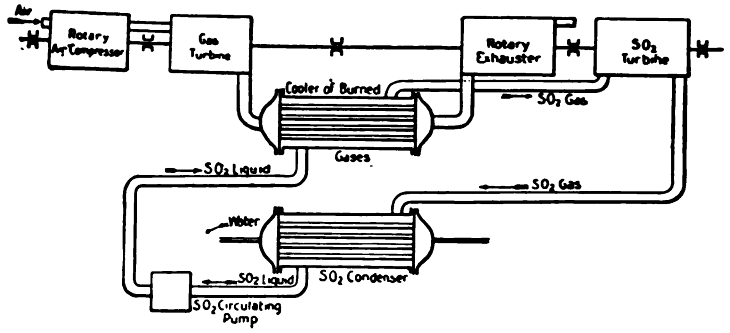


FIG. 55.—Turbine with exhaust at reduced pressure, with recovery of waste heat by a sulphur dioxide turbine.

The system of exhausting at low pressure enables a regenerator to be employed, but the construction of the regenerator is not very easy, because the transmission of heat is not very active at low pressures.

There are three methods in which the system may be employed.

The temperature of the exhaust may be brought below 700 degrees by the use of some one of the cycles already discussed, this plan permitting the use of a multistage turbine, although at the expense of a certain increase in the work of compression. The actual balance of power can only be definitely determined by examining each case by itself. Still we have already seen, that, in general, if we attempt to increase the efficiency η by the use of a multistage turbine, the total effect will be improved to a greater extent if we increase the amount of heat supplied rather than by lowering the temperature of the exhaust.

The second application of the system which we are considering consists in increasing the expansion ratio, and utilizing this increase to allow a corresponding increase in the amount of heat supplied, while maintaining the temperature of the exhaust at 700 degrees. This method presents especial advantages in cases in which the efficiency is limited by consideration of the maximum pressure of the cycle; which is notably true in the explosion cycles.

Finally, we may utilize the low-pressure exhaust in a manner which avoids the use of a piston compressor. We shall see that multicellular turbine compressors are not well adapted for the production of very high pressures. Under such conditions their efficiency is materially reduced by reason of the friction of the latter wheels of the series in the gas or air of high density. It is therefore better to arrange a turbine compressor to deliver the gas into the combustion chamber at a pressure, say, of six atmospheres, and follow the gas turbine by an exhaust blower, reducing the exhaust pressure to $\frac{1}{6}$ atmosphere, than it is to employ a single compressor operating at a pressure of 36 atmospheres.

In addition, the power turbine will operate with less frictional resistance by reason of the lower pressure.

It seems as if some such arrangement as this is necessary if the piston compressor is to be entirely eliminated in gas turbine design.

From a thermodynamic point of view there should be no difference between the operation with exhaust at low pressure or at atmospheric pressure, provided the ratio of the two extreme pressures is the same in both cases; and provided that the two compressors operate isothermically and at the same temperature T_0, in both cases.

III.

Cycles Using Heat Regenerators.

It is understood that it is possible to employ cycles having the same efficiency as that of Carnot between the same limits of temperature, by replacing the adiabatics of the Carnot cycle by two isodiabatics. The two simplest solutions of this problem are those of Stirling and of Ericsson, but the first of these involves reheating under constant volume, and is not applicable to our case.

The Ericsson Cycle.

In the Ericsson cycle, on the contrary, the exchanges of heat are made under constant pressure: the two isodiabatics are isobarics.

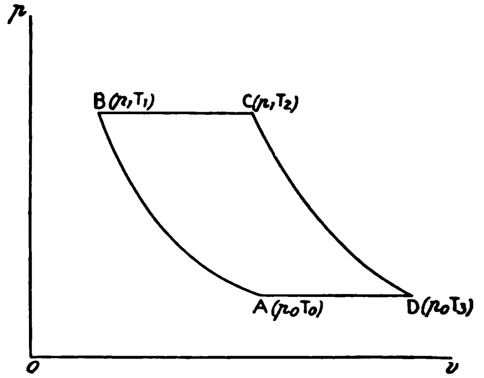


FIG. 56.—Ericsson cycle.

The gas is compressed along AB at the constant temperature T_0; it is reheated under constant pressure (p_1) along BC, by means of a regenerator, which raises the temperature to T_2. The heat furnished by the fuel is introduced along CD at the constant temperature T_2, during which the pressure falls to p_0. Finally the gas is cooled in the regenerator, from T_2 to T_0, at the constant pressure p_0.

Unfortunately this cycle cannot be realized in practice any more than can the Carnot cycle.

Independently of the practical difficulty of obtaining an isothermal combustion in the expansion nozzle, we encounter the impossibility of introducing large quantities of heat without using excessively high compressions, for we have:

$$\frac{p_1}{p_0}=e^{\frac{Q}{ART_2}}.$$

The thermal efficiency $\rho=1-\frac{T_0}{T_2}$ cannot exceed 0.57, since the gases are discharged upon the turbine wheel at a temperature T_2.

The work of compression $\mathfrak{T}c$ is given by:

$$RT_0 \log \text{hyp}\left(\frac{p_1}{p_0}\right) \quad \text{or} \quad RT_0\frac{Q}{ART_2}$$

whence
$$\frac{\mathfrak{T}c}{\mathfrak{T}u}=\frac{EQ\frac{T_0}{T_2}}{EQ\left(1-\frac{T_2}{T_0}\right)}=\frac{1}{\frac{T_0}{T_2}-1}.$$

We then have $\rho=0.57$ and $\frac{\mathfrak{T}c}{\mathfrak{T}u}=0.75$

also $\eta=0.15$ and $\rho\eta=0.15\times 0.57=0.086$

We see, therefore, that the Ericsson cycle is neither practicable nor advantageous. It is possible, however, to apply the principle of regeneration to other cycles, and as we shall see, with advantageous results.

In general, the method of regeneration is available only for cycles using isothermal compression, and especially those in which the combustion takes place under constant pressure.

Cycles Employing Isobaric Introduction of Heat.

As large a proportion as possible of the heat contained in the exhaust gases should be recovered by passing these hot gases through a system of tubes by means of which they heat the compressed gas on its way to the combustion chamber. We see that with a regenerating surface of infinitely great extent we might recover all the heat, if the compressed gas to be heated left the compressor at the ordinary temperature (say about 300° C. absolute). This involves an isothermal compression, while if the compression is adiabatic the exhaust gases cannot be cooled below the final temperature of compression.

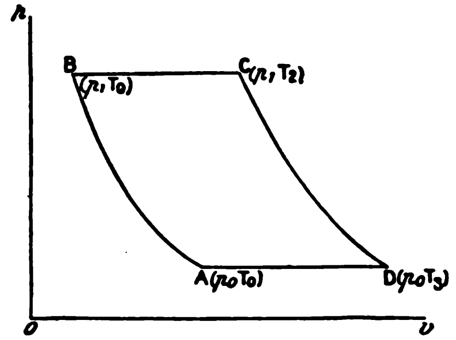


FIG. 57.—Cycle with isobaric combustion and isothermal compression.

The compression is accompanied by a consumption of heat:

$$Q' = ART_0 \log \text{hyp} \left(\frac{p_1}{p_0}\right).$$

We then introduce by regeneration K calories under the constant pressure p_1, and the temperature passes from T_0 to T_1. We have

$$K = C_p(T_1 - T_0).$$

The fuel, furnishing Q calories, raises the temperature from T_1 to T_2:

$$Q = C_p(T_2 - T_1).$$

The adiabatic expansion from p_1T_2 to p_0T_3 gives:

$$\frac{T_3}{T_2}=\left(\frac{p_0}{p_1}\right)^{\frac{\gamma-1}{\gamma}}.$$

If the regeneration could be complete the gas would enter the regenerator at T_3 and leave it at T_0, the surrounding temperature, leaving behind it $C_p(T_3—T_0)$ calories. But in reality the temperature of the gases is not reduced to T_0, besides which the compressed gas cannot acquire all the heat units thus gathered, because of the losses by radiation, conductivity, etc. If we call the total efficiency of the operation μ, we have:

$$K=C_p(T_1-T_0)=\mu(T_3-T_0).$$

For example, if the gases leave the turbine at 700 degrees they contain 92 calories per kilogramme which are recoverable, and we have $K=92\ \mu$.

Here the quantity of heat introduced in the cycle is $(K+Q)$, the quantity given up in cooling during the compression is Q', and that which is discharged with the exhaust is equal to

$$q=C_p(T_3—T_0).$$

The quantity of heat converted into useful work is therefore equal to:

$$(K+Q)-Q'-C_p(T_3-T_0).$$

The actual amount of heat abstracted from the fuel being Q, we then have:

$$\rho=\frac{(K+Q)-Q'-C_p(T_3-T_0)}{Q}.$$

If we maintain a standard temperature of the exhaust, say 700° C. absolute, the value $(K+Q)$ of the total heat introduced is equal, for each compression ratio, to that which has been computed for cycles without regeneration. It follows that the useful work obtained per kilogramme of air

Cycles Using Isothermal Compression with Isobaric Introduction of Heat and Regeneration.

Ratio of Compression $\frac{p_1}{p_0}$	5	10	15	20	25	30	40	60	80	100
Temperature of combustion T_2	1120	1365	1533	1680	1780	1880	2050	2300	2500	2670
Total introduction of heat $K + Q$	197	256	292	328	350	375	415	480	522	563
Equivalent $A\mathfrak{T}_c$ of work of compression	33	48	56	62	67	71	76	85	91	95
Equivalent $A\mathfrak{T}_u$ of useful work	70	114	144	174	191	212	247	303	339	377
Ratio $\frac{\mathfrak{T}_c}{\mathfrak{T}_u}$ of work of compression to useful work	0.47	0.42	0.39	0.36	0.35	0.34	0.31	0.28	0.26	0.25
Mechanical efficiency η	0.36	0.39	0.42	0.44	0.445	0.45	0.47	0.49	0.51	0.52
Heat abstracted from the combustible Q, calories: $\mu = 0.50$, $K = 46$	151	210	246	282	304	329	369	434	476	517
$\mu = 0.75$, $K = 69$	128	187	223	259	281	306	346	411	453	494
$\mu = 1.00$, $K = 92$	105	164	200	236	258	283	323	388	430	471
Thermal efficiency ρ, for $\mu = 0.00$, $K = 0$	0.36	0.45	0.49	0.53	0.545	0.565	0.595	0.63	0.65	0.670
$\mu = 0.50$, $K = 46$	0.465	0.54	0.585	0.615	0.63	0.645	0.67	0.70	0.71	0.730
$\mu = 0.75$, $K = 69$	0.55	0.61	0.64	0.67	0.68	0.70	0.71	0.74	0.75	0.760
$\mu = 1.00$, $K = 92$	0.665	0.70	0.72	0.735	0.74	0.75	0.765	0.78	0.79	0.800
Total useful effect $\rho\eta$ for $\mu = 0.00$, $K = 0$	0.130	0.180	0.205	0.233	0.243	0.255	0.280	0.310	0.330	0.348
$\mu = 0.50$, $K = 46$	0.167	0.210	0.245	0.270	0.280	0.290	0.315	0.342	0.360	0.380
$\mu = 0.75$, $K = 69$	0.198	0.238	0.270	0.235	0.300	0.315	0.335	0.363	0.380	0.395
$\mu = 1.00$, $K = 92$	0.239	0.273	0.300	0.323	0.330	0.337	0.360	0.382	0.402	0.415
Equivalent of net mechanical work available	25	45	60	77	85	96	116	148	173	195
Consumption of air: kg. per H. P. hour	25.5	14.3	10.6	8.3	7.5	6.7	5.5	4.3	3.7	3.3

is not increased by the use of the regenerator, and the ratio of the work of compression to the useful work is not changed.

The table on page 146 gives the results for various ratios of compression, and for an exhaust temperature of 700° C. absolute.

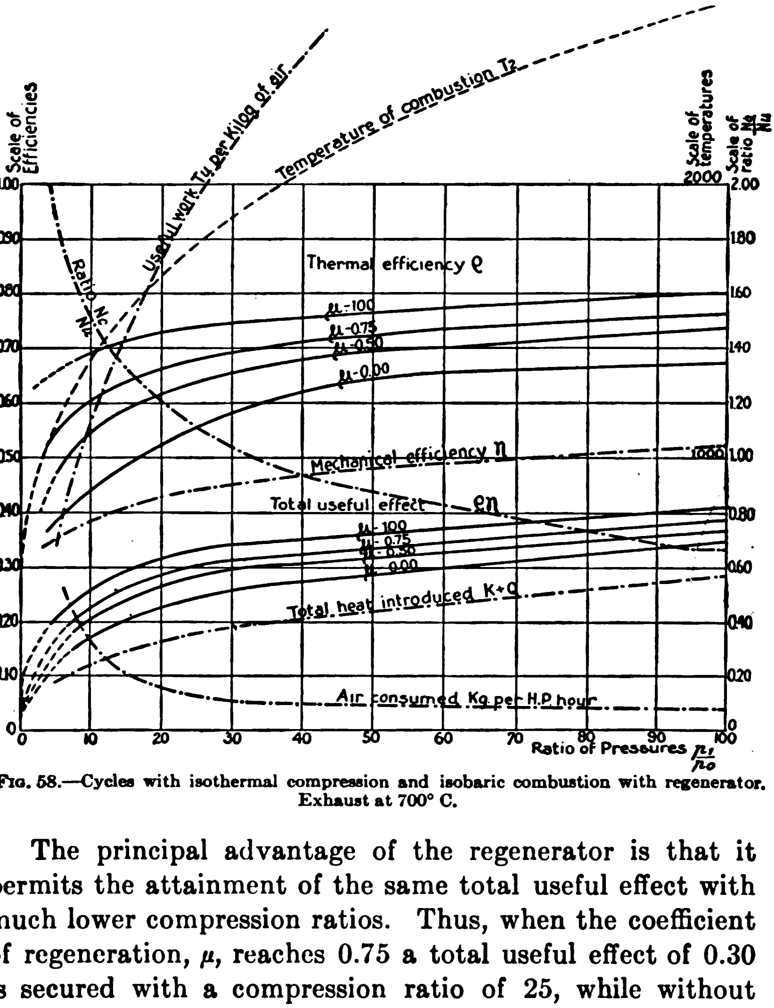


FIG. 58.—Cycles with isothermal compression and isobaric combustion with regenerator. Exhaust at 700° C.

The principal advantage of the regenerator is that it permits the attainment of the same total useful effect with much lower compression ratios. Thus, when the coefficient of regeneration, μ, reaches 0.75 a total useful effect of 0.30 is secured with a compression ratio of 25, while without regeneration a compression of 60 would be required to secure the same efficiency.

For a given compression, a regeneration of 50 per cent. ($\mu=0.5$) improves the total useful effect $\rho\eta$ from 10 to 30 per cent. and a regeneration of 75 per cent. improves the total useful effect from 15 to 30 per cent., according to the compression; the influence of the regenerator being greater with low compressions than with the higher compression ratios.

The use of the regenerator does not extend the limit of maximum total useful effect except when the compression ratio exceeds 80. In this case we are limited by the calorific value of the fuels, which control the maximum amount of heat introduced. The use of the regenerator permits this maximum to be extended.

Summing up, with a coefficient of regeneration of 0.75, which seems reasonable, we have theoretically, for a compression ratio of 25, the possibility of obtaining an effective horse-power hour with 2100 calories; and with a compression ratio of 60 we may produce a horse-power hour with about 1800 calories. These are very encouraging results.

The following table (page 149) gives the quantity of heat transmitted to the regenerator to obtain a given amount of available mechanical work.

It is seen that high compressions have the advantage of reducing, to a large degree, the quantity of heat to be regenerated. Thus, with a value $\mu=0.75$ it is necessary to transmit to the regenerator 510 calories per net effective horse-power if the compression ratio is 25, and only 290 calories if the compression ratio is 60.

The transmission of heat per effective horse-power in the condensers of steam engines ranges from 3180 to 6360 calories per horse-power hour, according as the engine consumes 5 or 10 kilogrammes of steam per horse-power, while the condensing surface ranges from 0.10 to 0.30 square metre per horse-power.

The exhaust gases of the turbine enter the regenerator at a temperature of 700° C. absolute, and should leave it at

Table of Regenerator Action.

Ratio of compression $\frac{p_0}{p_1}$		5	10	15	20	25	30	40	60	80	100
Ratio of heat regenerated to useful work $\frac{K}{A\mathfrak{C}u}$	$\mu=0.50$ $K=46$	0.66	0.40	0.32	0.265	0.24	0.215	0.185	0.15	0.14	0.12
	$\mu=0.75$ $K=69$	0.99	0.60	0.48	0.40	0.36	0.33	0.28	0.23	0.21	0.18
	$\mu=1.00$ $K=92$	1.32	0.80	0.64	0.53	0.48	0.43	0.37	0.30	0.27	0.24
Ratio of heat regenerated to net effective work $\frac{K}{A\eta\mathfrak{C}u}$	$\mu=0.50$ $K=46$	1.84	1.02	0.77	0.60	0.54	0.48	0.395	0.31	0.27	0.24
	$\mu=0.75$ $K=69$	2.76	1.53	1.16	0.90	0.81	0.72	0.60	0.46	0.40	0.35
	$\mu=1.00$ $K=92$	3.68	2.04	1.54	1.20	1.08	0.96	0.79	0.62	0.53	0.47

400° absolute to realize a regenerative efficiency of 75 per cent. The compressed air enters at 300° and leaves at 600°. With a systematic circulation the drop in temperature will be 100° from the entrance to the exit, and the mean temperature drop

$$\delta = \frac{D-d}{\log \text{hyp}\left(\frac{D}{d}\right)}$$

is thus 100 degrees.

We find in steam superheaters a heat transmission of 10 to 15 calories per hour, per square metre, per degree, or in the present case, 1000 to 1500 calories.

We then have:

$$\frac{510}{1000 \text{ to } 1500} = 0.34 \text{ to } 0.51$$

square metres, or

$$\frac{290}{1000 \text{ to } 1500} = 0.20 \text{ to } 1.30$$

square metres per net effective horse-power.

Thus, to realize a regenerative efficiency of 75 per cent. with a compression ratio of 60, it will suffice to have about the same area of heat transmitting surface as would be given to the condenser surface for a steam engine of the same effective power.

Regeneration in Cycles Using the Isopleric Introduction of Heat.

This case may be treated in the same manner as the preceding. It is necessary, however, to note that in practice the regenerated heat can be introduced only at constant pressure and does not act to raise the final pressure. The latter will, therefore, not be so high for a given initial compression as in cases in which there is no regeneration, and this diminishes somewhat the advantages of the explosion cycle.

The following results are deduced from the computation:

$$\rho = \frac{Q - \left(\frac{1}{\mu} - 1\right)K - ART_0 \log \text{hyp}\left(\frac{p_1}{p_0}\right)}{Q}$$

$$T_2 = \left[\frac{C_p}{K + C_p T_0} \frac{p_1}{p_0}\right]^{\gamma - 1} \times T_3{}^{\gamma}$$

which give for $\mu = 0.5$ and $T_3 = 700°$ the results in the following table.

Compression ratio $\frac{p_1}{p_0}$	5	10	15	20
Temperature of explosion T_2	1540	2030	2380	2680
Final pressure of explosion p_2	16	41	73	110
Heat furnished by combustible Q	177	260	319	370
Thermal efficiency ρ	0.55	0.64	0.68	0.71
Equivalent $A\mathfrak{T}u$ of useful work	98	166	217	262
Ratio $\frac{\mathfrak{T}c}{\mathfrak{T}u}$	0.33	0.29	0.26	0.24
Mechanical efficiency η	0.46	0.49	0.51	0.53
Total useful effect $\rho\eta$	0.25	0.31	0.35	0.38
Equivalent $A\eta\mathfrak{T}u$ of net mechanical work $\eta\mathfrak{T}u$	45	81	111	139
Consumption of air, kg. per H.P. hour eff.	14.10	7.90	5.70	4.60
Ratio of calories regenerated to effective work	1.00	0.57	0.42	0.33

We thus obtain the same results as with combustion at constant pressure, but with compressions only about one-half as great. The absolute maximum of useful effect is not increased, since we are limited by the consideration of the pressure and temperature of explosion to compression ratios only about one-half as great.

IV.

Cycles Involving the Injection of Water, Steam, or Cool Gases.

We have already seen that it is very desirable to be able to reduce the amount of gas to be compressed to realize a given amount of work. If, to fix our ideas upon this matter, we assume compression ratios above 80 to be excessive, we cannot introduce more than 450 calories per kilogramme of gas, while there are certain combustible mixtures which readily furnish from 550 to 600 calories. We are therefore obliged to dilute these latter, and thus increase the volume of gas to be compressed some 20 to 30 per cent. This inconvenience becomes aggravated with lower compressions.

This fact has led to investigations as to whether we may not use the rich combustible mixtures without dilution by using certain artifices to limit either the temperature of combustion or the terminal temperature.

Limitations of the Temperature of Combustion.

The external cooling of the combustion chamber is entirely inconvenient. The calories thus abstracted take no part in the development of power. It would be simpler and more economical to reduce the amount of combustible introduced.

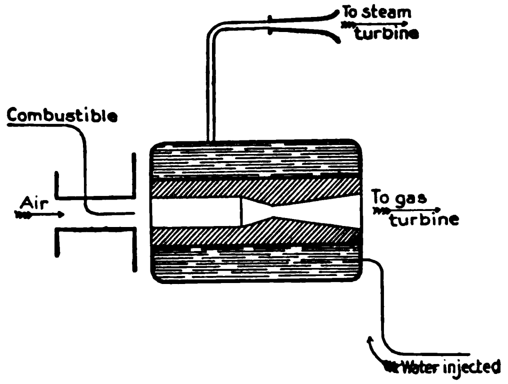


FIG. 59.—Combustion chamber for gas and steam turbines.

But, if the heat abstracted can be used to vaporize water, and if the steam thus produced is delivered, either to a separate turbine; or, by a separate nozzle, to the main gas turbine; or into the expansion nozzle of the gas; or, finally, into the combustion chamber itself, this heat will partake in the development of power according to a cycle more or less effective, and the loss will be reduced.

Suppose, for instance, that the steam thus produced is utilized in a separate turbine, which may be either connected to a condenser or exhaust into the atmosphere. The combustion chamber of the gas turbine will then act as the furnace of the steam boiler for the separate turbine. Leaving aside, for the moment, the complication of this arrangement, and assuming that we vaporize the water in the generator to a pressure of 20 atmospheres and superheat the steam to a temperature of 700 degrees absolute, by means of the calories derived from the walls of the combustion chamber we obtain a temperature of ebullition of 488 degrees absolute.

The heat contained in a kilogramme of water will be:

$$\lambda_{700} = 673 + 170 = 843 \text{ calories.}$$

If the exhaust is discharged into the air, the temperature will be 373° absolute, while if a condenser is used the temperature will be about 320°.

The thermal efficiency of the steam portion of the system will be:

Exhausting into atmosphere:

$$\rho = \frac{\lambda_{700} - \lambda_{373}}{\lambda_{700}} = \frac{843 - 637}{843} = 0.245$$

$$\rho\eta = 0.172$$

Exhausting into a condenser:

$$\rho = \frac{\lambda_{700} - \lambda_{320}}{\lambda_{700}} = \frac{843 - 620}{843} = 0.265$$

$$\rho\eta = 0.185.$$

In steam turbines the mechanical efficiency η is about 0.70. The result obtained when operated with a condenser corresponds to a consumption of steam of about 4 kilogrammes per effective horse-power (8.8 pounds). Now a gas turbine, without regeneration or water injection and with a compression ratio of 10, gives a total efficiency $\rho\eta = 0.18$.

The arrangement which we have been discussing is therefore without interest as regards efficiency unless we adopt compressions higher than 10. The only advantage lies in the reduction in the importance of the compressor.

If the steam, produced at the expense of the heat developed in the combustion chamber, is delivered upon the wheel of the gas turbine through separate nozzles, the efficiency will be the same as above, and the same conclusions

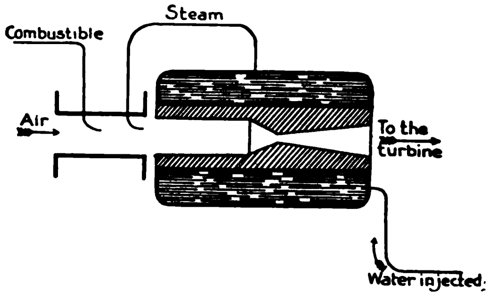


FIG. 60.—Combustion chamber for mixed turbine taking steam from jacket.

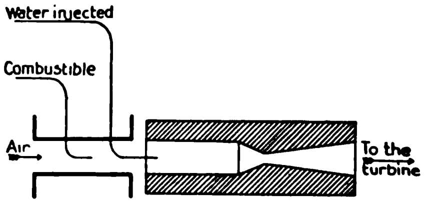


FIG. 61.—Combustion chamber for mixed turbine with independent water injection.

follow. The same is true if the steam is mingled with the burned gases in the expansion nozzles of the gas turbine itself; and with this arrangement certain precautions, important from a kinetic point of view, are necessary, as will be seen hereafter.

Finally, if the steam produced at the expense of the heat in the combustion chamber is delivered into the combustion chamber itself, the result will be the same as if the water were delivered directly into the combustion chamber in the liquid form. This is the arrangement which we shall now examine (Fig. 60).

Let x be the weight of water injected per kilogramme of gas burned, and let p be the pressure in the combustion chamber. The tension p^1 of the steam is found from the law of the mixture of gases and vapors, and is equal to:

$$p^1 = p\frac{R^1 x}{R+R^1 x}$$

in which R and R^1 are the specific constants of air and of the vapor of water. Supplying these constants, we have:

$$p^1 = p\frac{46.8x}{29.3+46.8x}.$$

Let θ be the temperature of ebullition which corresponds to this pressure p^1.

The heat absorbed by the vaporization of 1 kilogramme of water injected at 0°, into the combustion chamber, is given by:

$$\lambda_\theta = q + r = 606.5 + 0.305(\theta - 273).$$

The steam produced is also superheated, and if we represent the mean value of the specific heat of this steam, superheated between the temperatures of θ and T_2, by $\overline{Cp_{\theta T_2}}$, the superheating will absorb

$$\overline{Cp_{\theta T_2}}(T_2 - \theta) \text{ calories.}$$

The total amount of heat absorbed by 1 kilogramme of steam may readily be calculated by assuming 0.48 as the mean value of the specific heat of steam, and by using the formula of Lorenz

$$\overline{Cp_{\theta T_2}} = a + \frac{T_2^3}{b}$$

which gives:

$$\lambda_{T_2} = \lambda_\theta + \int_\theta^{T_2}\left(a + b\frac{p}{T_2^3}\right)dT \quad = \lambda_\theta + a(T_2 - \theta) - \frac{bp}{2}\left(\frac{1}{T_2^2} - \frac{1}{\theta^2}\right)$$

with $\qquad a = 0.43$ and $b = 36 \times 10^5$.

If we take the value of γ the same for the superheated steam as for the gas, we may calculate the temperature at the end of the expansion T_3 and the corresponding heat of the steam λ_{T_3}, from whence the thermal efficiency ρ of the steam, considered separately, will be:

$$\rho = \frac{\lambda_{T_2} - \lambda_{T_3}}{\lambda_{T_2}}.$$

It will be observed that λ_{T_2} and λ_{T_3} are dependent upon the ratio x, or the proportion of water to gas, by weight. The lower this ratio is the more the tension of the steam is reduced with relation to the pressure of combustion p. If the computations are made it will be found that the results differ very little from those corresponding to the case of saturated steam without the presence of any air (in which x = infinity) at least when the temperatures are relatively high, as in the case which we are considering.

The following table gives the results:

Absolute pressure of combustion p		5	10	15	20	25	30	40
Temperature of combustion T_2		1120	1305	1533	1680	1780	1880	2050
Temperature of ebullition θ		425	453	472	488	498	503	523
Heat of vapor λ_{T_2}		990	1130	1230	1310	1390	1490	1580
Heat of vapor λ_{T_3}		790	790	790	790	790	790	790
Thermal efficiency ρ	for the vapor	0.20	0.30	0.36	0.40	0.43	0.47	0.50
	for the gas	0.34	0.43	0.47	0.52	0.55	0.57	0.60
Total efficiency $\eta\rho$	for the vapor	0.14	0.21	0.25	0.28	0.30	0.33	0.35
	for the gas	0.11	0.16	0.18	0.22	0.25	0.26	0.28

This table is computed on the assumption that x = infinity, $T_3 = 700°$ absolute, and that the exhaust is discharged against atmospheric pressure. If the exhaust pressure is reduced the figures will be modified.

It will be seen that the thermal efficiency of the cycle of the vapor is lower than that for the gas, but if we consider the total efficiency $\eta\rho$, taking the efficiency of the turbine at 0.7, and that of the compressor (including its transmission) also at 0.7, these results are reversed. This follows because the work of the compressor is reduced by the use of the steam.

We conclude from this analysis, that the injection of water is more advantageous than the introduction of an excess of air for combustion, above all because it permits a material reduction in the dimensions of the compressor.

It may be desirable to consider whether or not there is any risk of the dissociation of the water under the conditions of temperature and pressure existing in the combustion chamber. In all probability there is no danger of such action, since dissociation does not begin, at atmospheric pressure, until a temperature of 1300 degrees C. absolute, and the tension of dissociation does not reach a value of 0.5 until 2100° C. At the pressures under consideration there can therefore be no appreciable dissociation, and there can be still less during the expansion, for the drop in temperature with the pressure is very rapid.

Injection into the Combustion Chamber, of Steam Produced in a Regenerator.

It has been proposed to replace the introduction of an excess of air in the combustion chamber by an injection of steam. If this steam is produced by the combustion of fuel under a boiler the result will be the same as in the case of the injection of water which we have just examined. This arrangement, however, would be accompanied with the heat losses involved in the use of a separate boiler, together with the mechanical complications accompanying it, besides which it would be necessary to inject much more steam to produce the same effect.

Assuming, as before, that x = infinity, the total amount of heat absorbed by the injection of a given weight of water in the liquid state is 1.5 to 2.5 times greater than if it is injected in the form of steam at θ degrees.

The injection of steam into the combustion chamber is of interest only when the steam is generated in some form of regenerator, heated by the exhaust gases of the turbine. We will examine this case, always assuming that the pressure of the steam in the mixture is the same as that of the pressure of combustion.

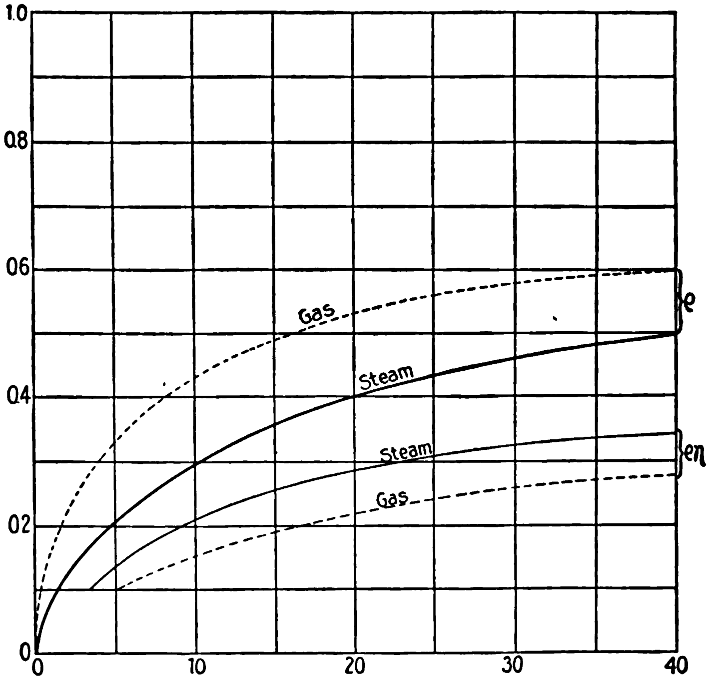


FIG. 62.—Efficiencies for mixed turbines.

The exhaust being at the temperature of 700 degrees absolute, each kilogramme of exhaust gases represents 92 calories. Each kilogramme of water carries 790 calories, but only 153 calories (the sensible heat) can be regenerated without condensation, if the exhaust takes place at atmospheric pressure. The result will be improved if the exhaust occurs at reduced pressure, and if we take into account the fact that the pressure of the vapor is lower than that of the surroundings into which it is discharged.

If, then, x kilogrammes of water are mingled with 1 kilogramme of burnt gases we may regenerate $92+153x$ calories, and the effective recuperation will be $\mu(92+153x)$, which gives a vaporization of a weight of water x:

$$\mu(92+153x)=(q+r)x$$

$$x=\frac{92}{\frac{q+r}{\mu}-153}.$$

The weight of steam which may be injected is thus well defined and distinctly limited. If θ is the temperature of ebullition corresponding to the pressure of the vapor of water in the mixture, each kilogramme of steam injected will absorb a quantity of heat equal to $\overline{Cp_{\theta T_2}}$ $(T_2-\theta)$. This quantity is computed below, assuming for simplification that θ is equal to the temperature of ebullition at the pressure p (page 160).

The injection of water absorbing γ calories per kilogramme of gas burned, it is possible to increase the amount of heat introduced by an equal amount without modifying the temperatures T_2 and T_3, provided the calorific power of the combustible will permit it.

It will thus be found, if we take the same efficiency for the regenerator (0.75, for example), that the total useful effect obtained differs very little from that secured by the use of a regenerator heating the compressed air. Nevertheless the actual consumption of air per effective horsepower is less, a fact which has a distinct practical advantage.

This method is especially applicable when the nature of the combustible permits the introduction of a large amount of heat, and when the exhaust is discharged at a reduced pressure.

Cycle with Isothermal Compression and Isobaric Combustion with Injection of Steam Generated by Waste Heat. Exhaust Temperature 700° Absolute.

Pressure of combustion p_1	5	10	15	20	25	30	40
Temperature of combustion........degrees	1120	1365	1533	1680	1780	1880	2050
Heat of vapor $q+r$........calories	653	662	668	673	676	677	683
Weight of vapor x for $\mu=0.75$........kilogrammes	0.127	0.126	0.125	0.124	0.123	0.122	0.121
Heat Q introduced per kg. of air........calories	189	245	282	317	350	375	415
Additional heat introduced by steam injection........calories	43	59	71	81	88	96	103
Total heat introduced........calories	232	304	353	398	433	471	524
Heat carried off by gases........calories	92	92	92	92	92	92	92
Heat carried off by the steam........calories	19.5	19.3	19.1	19	18.9	18.7	18.5
Heat carried off in the compressor........calories	33	48	56	62	67	71	76
Equivalent of useful work $\mathfrak{T}u$........	87	145	186	225	260	289	337
Thermal efficiency ρ........	0.375	0.475	0.525	0.565	0.575	0.615	0.640
Mechanical efficiency η........	0.43	0.46	0.48	0.50	0.51	0.52	0.536
Total useful effect $\rho\eta$........	0.16	0.22	0.25	0.28	0.31	0.32	0.34
Net effective work per kg. of air $\eta\mathfrak{T}u$........	38	67	90	112	132	150	180
Ratio $\frac{\mathfrak{T}c}{\mathfrak{T}u}$........	0.38	0.33	0.30	0.28	0.26	0.25	0.23
Ratio of powers $\frac{Nc}{Nu}$........	1.26	1.03	0.90	0.80	0.72	0.68	0.60
Consumption of air per H.P. hour........	17.0	9.5	7.1	5.7	4.8	4.3	3.5

The regenerator may be made in the form of a boiler imilar to the Serpollet flash boiler, or of the type proposed y Colonel Renard.

The gases enter the regenerator at a temperature of 700 egrees, and leave it at about 400 degrees absolute. The rater, raised from a temperature of zero to 450 or 500 egrees absolute, will be vaporized at this latter temperaure, at a pressure of about 5.30 atmospheres. It is easy o compute that the mean drop in temperature will be about 00 degrees in the boiler, and 75 degrees in the regenerator, orresponding to a heat transmission of 3000 and 1500 alories respectively per square metre per hour. This will equire about 0.0366 square metre of surface per kilogramme f air consumed per hour in the turbine, or about 0.16 square ıetre per horse-power delivered on the shaft, the consumpion per horse-power hour being 4.25 kilogrammes of air, nd 0.51 kilogramme of water, for a combustion pressure f 30 atmospheres.

The necessary heating surface will therefore be of the ame order of magnitude as that of the condenser of an ordiary marine engine, but probably greater than that of a reenerator for a gas turbine using a regeneration of gas to gas.

Practically, vaporization under pressures exceeding 0 atmospheres may appear to offer certain difficulties. 'his method of regeneration, however, becomes very simple the exhaust is discharged at reduced pressure. The presure of combustion, for example, being from 5 to 10 atmosheres, and the exhaust pressure $\frac{1}{5}$ atmosphere. The regenrator-boiler should be operated at pressure ranging only rom 5 to 10 atmospheres. The drop in temperature would e materially increased, which would facilitate the transıission of heat. At the same time, the efficiency of the cycle rould be increased.

Under such a system, using a producer of the Gardie ype, operating under 5 to 10 atmospheres pressure, the loss of the sensible heat of the gas could be avoided, and the proportion of steam or water injected increased.

It may be noted that in the case of compressors using water injection, the vapor produced from the injected water is evolved with the gaseous mass, and permits an increase in the amount of heat introduced, thus improving the efficiency.

The Use of Large Injections of Water in Connection with a Very Rich Fuel. Turbines Using Liquid Oxygen.

As a matter of curiosity it may be noted that if pure oxygen be used in the combustion, the total weight of gas burned would be only about one-fourth that otherwise required; and therefore, the introduction of heat being quadrupled, might reach 2000 calories per kilogramme. The injection of water into the combustion chamber might then be materially increased. Such a mixed turbine would require a much smaller compressor, consuming much less power, or if liquid oxygen were used a small centrifugal pump operating at high pressure would replace the air compressor.

A machine of this kind would require three such pumps; one for the liquid oxygen, one for the liquid fuel, and the third for the water. A tubular heater, heated by the exhaust gases, would heat the water and vaporize the liquid oxygen, the only other elements required being the combustion chamber and the turbine wheel.

The temperature of combustion would be the same as before, but the temperature of the exhaust would be materially lowered by reason of the calories absorbed by the vaporization of the oxygen, so that the thermal efficiency should be at least equal to that computed above.

With regard to the mechanical efficiency η, this, neglecting the work absorbed by the pumps, would be above 0.70, because of the absence of the compressor. The total useful effect, $\rho\eta$, would therefore be 0.70 or 0.75 times 0.70, or about 50 per cent. Although the amount of work available would thus be very high, the velocity of discharge of the mixture would be much greater than in the ordinary case, and the mechanical efficiency of the turbine would be lower.

Such a machine, however, would be extremely light. It is true that it would be necessary to carry 4 kilogrammes of liquid oxygen and 5 kilogrammes of water for every kilogramme of petrol, but for certain applications the final result would be very favorable.

While this application of the gas turbine is yet within the domain of scientific curiosities, it is by no means an absurdity. M. Cailletet has not hesitated to propose a similar combination, using piston engines, for the design of extremely light and powerful motors for aerial or submarine navigation.

Limitations of the Temperature of Expansion. Injection of Water, Steam, or Cool Gases after Expansion.

If the exterior of the expansion nozzle is cooled, the expansion is no longer adiabatic, and cannot be subjected to computation. All the energy thus abstracted, however, is evidently lost.

The same is not the case if the expanding gases are cooled by an injection of water, since the vapor thus formed is added to the fluid mass. Nevertheless, at a temperature of 700 degrees, and at atmospheric pressure, about $\frac{8}{10}$ of the calories absorbed by the injection are lost and absorbed by the vaporization properly so-called.

We are therefore led to consider the injection of steam. If the velocity of the steam is lower than that of the current of gases, there is caused, as we shall see, an important loss of energy. Let us then assume that the two currents have the same velocity. In order to accomplish this, it is necessary that the vapor be generated at a pressure higher than that of the gas in the combustion chamber. We will pass over this difficulty. In order to obtain a better result than is secured by the direct injection of water the steam must be regenerated by the use of waste heat. Under these conditions, and assuming that the expanded steam is still saturated, dry, or slightly superheated, and calling x the weight of this steam delivered for each kilogramme of air, calculated as heretofore, we may complete the temperature T_3' of the expanded gas.

$$Cp(T_3'-700)=0.48(T_3'-373)x$$

whence
$$T_3'=\frac{167-180x}{0.24-0.48x}.$$

We may then calculate the new temperature of combustion, the new introduction of heat, and the new efficiency:

$$\rho'=\frac{Q'-(150\,x+92+A\mathfrak{T}_c)}{Q'}.$$

With a coefficient of regeneration of 0.75 we may inject 12 to 13 per cent. of water, and permit a final temperature of expansion of the gas of 800 degrees absolute instead of 700° C.

It is thus seen that for a given compression ratio, the useful effect is slightly lower than that obtained by injecting the steam before the expansion. In practice the injection of steam after the expansion offers considerable difficulties of a kinetic order.

We will now consider the injection of cool gases.

Injection of Cool Gases at Low Velocities.

Stodola has shown in the following manner that a mixture of two currents of gases having two different velocities w_1 and w_2 results in a material loss of kinetic energy.

There are two cases to be considered. The first corresponds to the use of a mixing chamber so formed as to permit the operation to be effected without raising the pressure.

The second case corresponds to the use of a cylindrical chamber, which leads to an elevation in the final pressure.

If we call dP_x the force acting axially upon an element dm, and call Π_1, Π_2, and $\Pi = \Pi_1 + \Pi_2$ the flow by weight, the theorem of quantities of motion gives:

$$\frac{ndt}{g}w - \left(\frac{n_1dt}{g}w_1 + \frac{n_2dt}{g}w_2\right) = \Sigma dtdP_x$$

from which we get:

$$\Pi w = \Pi_1 w_1 + \Pi_2 w_2.$$

This is the formula for impact of non-elastic bodies, and the loss of energy is:

$$Z = \frac{1}{2}\left(\frac{\Pi_1}{g}w_1^2 + \frac{\Pi_2}{g}w_2^2\right) - \frac{1}{2}\frac{\Pi}{g}w^2.$$

If we call T_3 and θ the respective temperatures of the two gaseous currents before mixture, and T_3' the temperature after mixture, we have:

$$\Pi_1(T_3 - T_3')Cp + \frac{Z}{A} = \Pi_2(T_3' - \theta Cp).$$

These three relations enable us to compute the temperatures and the efficiency.

Let us take the extreme case in which the gas is injected cold and without velocity. We have:

$$w_2 = O; \quad \text{and} \quad w = \frac{\Pi_1}{\Pi}w_1$$

whence

$$Z = \frac{\Pi_1\Pi_2}{\Pi}\frac{w_1^2}{2g}.$$

The ratio ε of the lost energy to the amount of energy available in the gaseous current before the mixture will then be, for this particular case·

$$\varepsilon = \frac{\Pi_2}{\Pi}.$$

For example, if $\Pi_2 = 1$ kilogramme, and it is desired to reduce the temperature to $T'_3 = 700$ degrees by injecting

Π_2 kilogrammes of air without velocity at 300 degrees absolute (or $\theta = 300$), the limiting case corresponding to $T_3 = T'_3$ will be attained when $\frac{Z}{A} = 95\ \Pi_2$.

If χ be the thermal equivalent of the kinetic energy of 1 kilogramme of burned gas before the mixture, we have: $\frac{Z}{A} = \frac{\Pi_1}{\Pi}\chi$, whence $\frac{\Pi_2}{\Pi}\chi = 95\Pi_2$, and $\Pi = \frac{\chi}{95} - 1$

For example,

for	$\chi = 100$	200	300	400 calories
we have	$\Pi_2 \geqslant 0.05$	1.10	2.15	3.20 calories
and	$\varepsilon \geqslant 0.05$	0.52	0.68	0.76 calories.

There is, therefore, a considerable loss in the kinetic energy of the gaseous current when the latter attains a considerable value.

Suppose that we are using a cylindrical mixing chamber. The pressure beyond the zone of mixture will then be higher than that in front of it. Professor Stodola, who has examined this question, finds that there may be two solutions, and that the velocity of the mixture may have two distinct values. One of these corresponds to a simple mixture, with a loss of kinetic energy and a relatively moderate rise in temperature. The other corresponds to a velocity greater than that of sound and involves the existence of a shock of compression of which we shall speak hereafter. However the mixture may be effected there is no more advantageous result to be expected than in the preceding case, and there is nothing to be deduced from the idea other than the results involved in progressive mixtures in successive chambers.

Injection of Cold Gases at the Same Velocity as the Principal Current.

Suppose now that the cold gases are given, by the use of a blower or similar apparatus, a velocity equal to that of the principal current, and that the mixing chamber is of such a shape that there is no increase in pressure. In this case there is theoretically no loss of energy.

It is, however, necessary to expend, in driving the blower, an amount of energy equal to the necessary kinetic energy. The question then presents itself as follows: Is it more advantageous to compress all the air required and deliver it at once to the combustion chamber, or to compress only a portion of it to the pressure of combustion, and to cause the remainder to be delivered by a blower to the mixing chamber?

Let us suppose, for example, that we have a combustible capable of permitting an introduction of about 520 calories per kilogramme of mixture. If we compress to 10 kilogrammes per square centimetre, we can introduce only about 260 calories per kilogramme to exhaust at 700 degrees. If we do introduce 520 calories we shall have a temperature of combustion of 2500 degrees and an exhaust of 1270 degrees. The kinetic energy will be equivalent to 290 calories per kilogramme of gas. In order to bring the temperature of 1270 to 700 it will be necessary to mix with each kilogramme of burned gases 1.32 kilogramme of air, and the kinetic energy to be imparted to this cold air will be equivalent to $1.32 \times 290 = 410$ calories. We then have at the outlet of the mixing chamber a kinetic energy equivalent to $290 + 410 = 700$ calories for 2.30 kilogrammes of mixture. We will get in work on the shaft $0.7 \times 700 = 490$. Now, the work required for the compressor will be 48 calories and for the blower 410, a total of 458. There will therefore each have to give an efficiency of 0.94 in order that they should not absorb more power than the turbine itself produces. The injection of cold gases is therefore wholly impracticable.

V.

Combination Cycles. The Adaptation of a Second Engine to Utilize the Waste Heat.

It has been suggested that the waste heat discharged by a gas turbine should be utilized to operate a second turbine, employing sulphurous acid gas, for instance, or even vapor of water. We have already shown that the exhaust gases should be discharged at a temperature of about 700 degrees C., absolute, and that it is not advantageous to lower this temperature by diminishing the amount of heat introduced.

The heat abstracted from the gas during compression may be carried off by water circulating about the compression cylinders and through the inter-coolers. The amount of heat thus withdrawn compares in importance with that escaping with the exhaust gases, but its temperature is much lower. Theoretically, the temperature of the jacket water should not materially exceed that of the atmosphere, and in no case should it be higher than 50 to 100 degrees C. It is therefore necessary to resort to some substance having a low boiling point, such as sulphurous acid, in order to utilize this heat in a secondary engine.

We have at our disposal from 150 to 200 calories per kilogramme of gas burned. If we use a steam turbine as the secondary motor, operating at a pressure of 20 kilogrammes per square centimetre, superheating to 700 degrees absolute, and operating condensing, the thermal efficiency will be:

$$\rho=\frac{\lambda_{700}-\lambda_{320}}{\lambda_{700}}=\frac{843-620}{843}=0.265^*.$$

The total useful effect will then be:

$$\rho\eta=0.70\times0.265=0.185.$$

Now if the vapor of water had been used directly with the gases of combustion it would have given a useful effect of about 0.30.

It has been proposed to replace the vapor of water by a gas which is readily liquefiable, such as sulphurous acid. According to Professor Josse an indicated horse-power may be obtained in the secondary motor with a consumption

* This result agrees with practice, since it corresponds to a consumption of 4 kg. (8.8 pounds) of steam per horse-power hour, and consumptions below 4.6 kg. have already been obtained.

of 7800 calories, or even with 5000 calories when operating at a pressure of 25 atmospheres (90 degrees C.) in the condenser.

This last figure gives a thermal efficiency of 0.127, or a net useful effect of 0.7×0.127, or 0.089.

This result might be materially improved if we could permit the superheating of the sulphurous acid gas without causing corrosion upon the parts of the turbine with which it came in contact.

In any case a secondary turbine would permit a recuperation of 150 to 200 calories $\times 0.09 = 14$ to 18 calories, if we use sulphurous acid, or $92 \times 0.185 = 17$ calories, if we use water, admitting a coefficient of recuperation μ equal to unity. Taking $\mu = 0.75$, we get work equal to about 13 calories per kilogramme of gas. Now the net mechanical effort $\eta \mathfrak{T}u$ realizable per kilogramme of gas, with or without recuperation, varies between 25 calories and 200 calories when the pressure of combustion varies from 5 to 100 atmospheres.

The amount of work recoverable by the use of a secondary machine is therefore not of sufficient importance to warrant the complication of a separate machine to secure it.*

VI.

Conclusions from the Thermodynamic Study of the Gas Turbine. Method of Development to be Adopted. Probable Efficiency. Probable Divergence Between Theory and Practice.

The study of the gas turbine from a thermodynamic point of view does not appear to reveal any combination capable of giving results greatly differing from those already obtained from the latest improved gas engines. The high thermal efficiencies theoretically probable seem to be offset by the low mechanical efficiency. But this latter is capable

* In practice the relative importance of the work recovered might be a little greater, since all losses of energy have the effect of increasing the amount of heat in the exhaust gases, and of such leaks we have taken no account.

of improvement, so that there remains a margin for progress which is encouraging for the future.

The analysis which we have undertaken may be reviewed as follows:

1. Combustion under constant volume, as compared with combustion under constant pressure, shows, for the same initial pressure, a better efficiency, while at the same

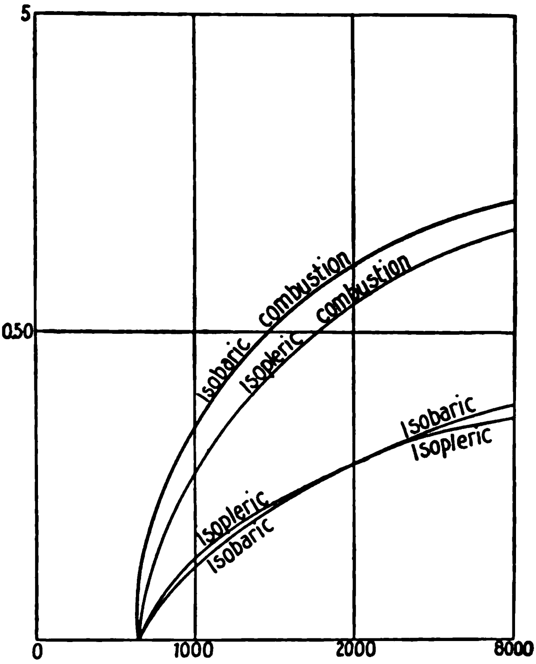


FIG. 63.—Efficiencies for various combustion temperatures.

time it permits the use of a less important compressor. Nevertheless, the absolute value of the efficiency is not greater, because we are more promptly limited by the maximum limit of permissible temperature of combustion T_2. This is true either for the specific power, or for the consumption of air per horse-power hour.

This method is advantageous, therefore, only from the point of view of the necessary compression ratio. This is a matter for consideration if we limit ourselves to the use of rotary compressors. In practice the mechanical efficiency is low because of the kinetic losses due to irregularities of flow under varying operation, besides the inconveniences attending an explosion machine. As a matter of fact, the explosion turbine is applicable only to very small powers, and for machines of light weight, in which the efficiency is a matter of secondary importance, and preliminary compression is undesirable.

2. Isothermal combustion involves excessively high ratios of compression, and is otherwise not practically realizable.

3. It follows that the best method available for the gas turbine corresponds to that for the gas engine; namely, combustion under constant pressure, with a preliminary isothermal compression.

4. If the ratio of the extremes of pressure has a given value, it is immaterial whether these pressures are high or low, in an absolute sense. This point is of interest in connection with the question of exhausting at low pressure, and with the use of multiple rotary compressors.

5. The temperature of the exhaust should be as high as practicable, with regard to the maintenance of the revolving wheels. It is deceptive to attempt to lower it by prolonging the expansion by the use of an air pump.

6. The best method of saving the heat escaping in the exhaust is by a simple tubular regenerator transferring the heat from outgoing to incoming gas. Regeneration by means of a steam boiler is worthy of consideration only for very rich combustibles, and the best plan then is to deliver the steam into a combustion chamber.

It may now be asked what important relations may be established practically between the above theoretical deductions and the practical results attainable with such machines as may actually be constructed.

It is probable that the practical cycles will differ from the theoretical ones in the gas turbine much as they do in the gas engine, but to a less extent.

Thus, as concerns the *compression*; there are two differences between theory and practice in effecting isothermal compression. One is the increase in the work of compression; the other, the elevation in temperature of the compressed gas, reducing the value of Q, and consequently the specific power. The results in practice lie between those computed for isothermal compression and those corresponding to adiabatic compression. But, as we have seen, the difference is not very great, and we have taken a sufficiently low value for the mechanical efficiency η_c to cover any discrepancy on this account.

With respect to the combustion, there are more important divergences between theory and practice, which must be taken into account. Thus, we have assumed that the reaction is effected in surroundings which are strictly adiabatic. This cannot be effected in practice, and notwithstanding all our precautions a loss of heat will occur.

The combustion will also be incomplete, hence there will be a loss of a portion of the combustible, or a partial dissociation, this being less probable under pressures of 30 to 40 atmospheres.

These three causes have one and the same result, an increase in the weight of combustible consumed per horsepower hour. This, however, does not affect the general development, especially if the fuel is in the liquid or solid state, since the additional amount of fuel required does not affect the work of compression.

The specific heat of the products of combustion differs materially from that of air, and varies with the temperature; and it is probable that the value taken for the temperature of combustion is greater than the real value. It follows that the introduction of heat is more limited than in our calculations, at least in the case in which this limit is fixed by the calorific value of the combustible. But since this limit is not likely to be reached in practice the only effect resulting from the disagreement between theory and practice is to reduce the amount of air required for dilution.

Finally, the expansion is not strictly adiabatic, and the form of the expansion curve is not precisely that which corresponds to the relation

$$pv^{1.41} = \text{constant}.$$

Thus, there is a loss of heat which may be small, but can never be strictly zero. Besides, the gas is heated to a certain extent by friction. The true law of the expansion can be determined only by experiment. In any case the exponent γ in the formula will differ from 1.41 because we are dealing with gases other than air and because the ratio $\frac{C}{c}$ is not constant when the temperature varies between very wide limits.

Even taking into account the variability of the specific heats with the temperatures, M. Vermand has shown that the law of Poisson is expressed practically by the relation:

$$pv^{\gamma} = \text{constant}$$

when $$\gamma = 1 + \frac{AR}{a}$$

in which $a = 0.162$, so that for air $\gamma = 1.441$.

If we use data obtained from certain trials of gas engines, we are led to accept for γ values such as 1.3 to 1.2.

The corresponding results differ materially from those which we have computed above.

Thus, to obtain the theoretical temperature of 700 degrees for the exhaust, with a combustion pressure of 30 atmospheres, it is necessary to produce a combustion temperature of 1880 degrees, introducing 375 calories per kilogramme of air.

These are the results obtained above for $\gamma = 1.4$ (giving $\rho = 0.57$).

If now, we take $\gamma = 1.3$, or $\gamma = 1.2$ we shall have:

$$\left(\text{since } \frac{T_2}{T_0} = 30^{\frac{\gamma-1}{\gamma}}\right)$$

$\frac{\gamma-1}{\gamma} = 0.23$	$\frac{\gamma-1}{\gamma} = 0.17$
$T_2 = 1526°$	$T_2 = 1253°$
$Q = 295$	$Q = 230$
$\rho = 0.45$	$\rho = 0.29.$

The efficiency is influenced, as we see, to a remarkable extent by the variation of γ; and the diagram (Fig. 64) shows

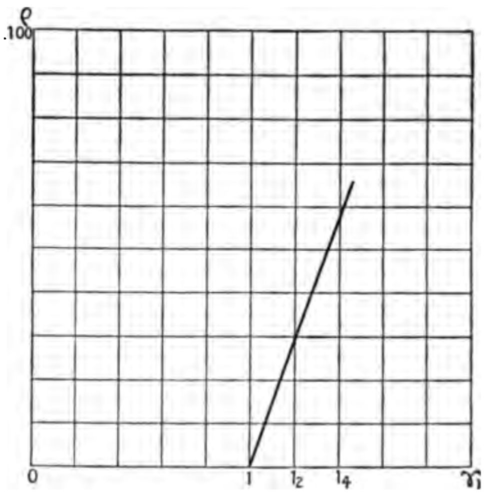


FIG. 64.—Variations of exponent of expansion.

that it varies almost in proportion to $\gamma - 1$, at least in the case under consideration, in which the cycle uses combustion at constant pressure and isothermal compression.*

* The thermal efficiency becomes zero for $\gamma = 1$, for we have $\frac{T_2}{T_0} = (30)^0 = 1$ and Q then becomes zero.

It may be of interest to note the following values for $\gamma = \frac{C}{c}$, at a temperature of zero, and at atmospheric pressure, for the gases named:

H, O, N, Air, CO	1.41
H_2O	1.34
CO_2	1.29.

In engines utilizing the explosion of gases behind a piston, the value of the exponent γ has been deduced from the form of the expansion curve, and the figures thus obtained range from 1.3 to 1.6. In such machines, however, the action of the walls of the cylinder play an important part, while in the diverging nozzle of a gas turbine this action is reduced to a minimum, because a continuous flow is maintained.

However this may be, the true law of expansion in the nozzle of a turbine constitutes the principal unknown practical element which presents itself in the gas turbine. It has even been maintained that γ may become equal to 1, and that the expansion may be accompanied by no drop in temperature, and hence be incapable of producing any useful effect.*

Some experimenters have not been able to find the drop in temperature by thermometric observations, and have attributed this fact to the heat developed by the friction of the gases upon the thermometer. We cannot go into this objection at length. The expansion doubtless follows the formula of Poisson, pv^γ = constant, but the true value of the exponent γ, and consequently of the efficiency, can be determined only by experiment.

We may note here a final reason for the discrepancies in our calculations between theory and practice. This is

* See Charles E. Lucke, Ph.D., Practical Investigations in the Gas Turbine Problem; Engineering Magazine, April, 1905. The Gas Turbine, Engineering Magazine, August, 1906.

the relative inexactness of the simple physical laws which we have accepted as relating to the substances under consideration: the laws of Mariotte, of Gay Lussac, of the constancy of specific heats, etc. In all standard works there may be found formulas which are more precise than those which we have used, and these may be substituted for the more simple laws. The greater degree of precision thus obtained is of minor interest, since the inevitable uncertainties of the question render any such excessive precision illusory.

Influence of the Nature of the Combustible.

Before leaving the thermodynamic study of the gas turbine it is desirable to examine whether the nature of the fuel available may have any important influence upon the possible efficiency.

In order to use the most advantageous cycles it is desirable, from what we have already seen, to be able to introduce from 375 to 415 calories, if we do not inject any steam, and from 470 to 524, if steam injection is to be used; the pressure ranging from 30 to 40 atmospheres.

Now, even using lean gases, such as that made in the Dowson producer, or the waste gases from blast furnaces, with a heating value of 800 calories per cubic metre (about 90 B.T.U. per cubic foot), it is possible to introduce about 460 calories per kilogramme of mixture; while with the richer gases, such as illuminating gas, acetylene, etc., we may get from 500 to 600 calories. The nature of the combustible has, therefore, a minor influence from this point of view. The constitution of the burned gases varies but little for the different combustibles, so that the specific heats are not greatly different. The cycles calculated upon the actual composition of the mixtures will therefore agree fairly well with those which we have based upon the properties of air.

It also follows that the total weight of gas to be compressed varies but little, and the same is true of the work required for compression. When a liquid fuel is used a greater amount of air is required per kilogramme of combustible than with a gaseous fuel.

It is an error to assume, as has sometimes been done, that gaseous fuels are less easily employed than liquid fuels. This may be the case for motors of the Diesel type because the intermittent action brings in the important question of ignition. Apart from the necessity for two separate compressors, however, the compression is not more troublesome when a gaseous fuel is employed. Since the combustion is continuous in the case of the gas turbine, the question of ignition is of secondary importance.

In the accompanying table the computations have been made according to the stated compositions of the various gaseous mixtures, taking the data calculated by M. Vermand.

It will be seen that C_p varies about 20 per cent., and γ about 1 per cent., in passing from one mixture to another.

The composition of the burned gases varies but slightly. Nitrogen predominates, being 62 to 74 per cent., followed by carbon dioxide, 12 to 33 per cent. Oxygen appears to be present in very small quantities, so that oxidation of metallic parts need hardly be feared.

The number of cubic metres of air and of gas to be compressed to correspond to the introduction of an amount of heat equal to 100 calories into the cycle, gives an idea of the necessary capacity for the compressor, or compressors. As this quantity varies from 148 to 180 litres, a difference of 22 per cent., this is not an element in which a serious error need enter.

In like manner the total weight of gas to be compressed per 100 calories introduced forms a measure of the total power required for the compression. The extreme limits are 0.17 and 0.22 kilogramme, a difference of 30 per cent.

Composition of mixture.	Thermal data.			Per cent. composition of gas burned, by weight.				Weight per litre of mixture burned 0° & 760 mm.
	Q	C	γ	H_2O	CO_2	N	O	
	calories							gramme
Illuminating gas + 6 vol. air	574	0.355	1.422	12.7	12.0	73.0	2.3	1.23
Dowson gas + 2 vol. air	463	0.334	1.334	9.5	23.5	66.5	0.5	1.38
Blast-furnace gas + 0.8 vol. air	456	0.312	1.419	4.6	33.0	62.0	0.4	1.40
Acetylene + 20 vol. air	625	0.302	1.433	3.1	14.9	74.0	8.7	1.31
Petrol, 10,000 calories + 23 kg. air	415	.....	.....	...	...	...	..	....

Composition of mixture.	Weight of cubic metre of burnt gas at 0° and 760 mm.	Weight of air per kilogramme of gas burned.	Volume of air per kilogramme of gas burned.	Volume of gas per kilogramme of gas burned.	Volume of air + volume of gas per kilogramme of gas burned.	For each 100 calories of heat introduced it is necessary to compress air + gas, total.	
	kg.	kg.	m^3	m^3	m^3	kg.	m^3
Illuminating gas + 6 vol. air	1.23	0.94	0.73	0.12	0.85	0.174	0.148
Dowson gas + 1.2 vol. air	1.38	0.59	0.46	0.38	0.84	0.215	0.181
Blast-furnace gas + 0.8 vol. air	1.40	0.44	0.34	0.42	0.76	0.220	0.166
Acetylene + 20 vol. air	1.31	0.96	0.74	0.04	0.78	0.192	0.148
Petrol, 10,000 calories + 23 kg. air	....	0.96	0.74	0	0.74	0.192	0.148

The lean gases are less advantageous in this respect than the richer mixtures.

We may therefore reach the conclusion that the nature of the combustible (liquid or gaseous) is not a matter of great importance from the point of view of the total probable efficiency $\rho\eta$, but that the lean mixtures are less advantageous than rich mixtures because of the unfavorable influence upon the mechanical efficiency when the work of compression acquires great importance.

The Gas Turbine from a Mechanical Viewpoint. Flow of Gases Through Nozzles.

According to the principle of the conservation of energy, the velocity of discharge w_2 is given if we neglect the initial velocity w_1 of the gas, by the relation:

$$\frac{w_2^2}{2g} = E(Q-q) = EC_p(T_2 - T_3) \qquad (1)$$

if the evolution undergone by the gas corresponds to an isothermal compression, followed by an isobaric combustion.

For the group of cycles exhausting at 700 degrees, we have:

Compression ratio.	5	10	15	20	25	30	40	60	80	100
$Q-q$ = calories.......	103	162	200	236	258	283	323	388	430	471
w_2 = metres per second	927	1162	1288	1400	1466	1533	1637	1794	1892	1975

These velocities exceed those of steam in ordinary steam turbines when compression ratios of 15 to 20 are exceeded. For this reason, under similar structural conditions for the revolving disc, the efficiency of the disc itself will be less. Fortunately other factors act in the opposite sense, and in favor of the gas turbine.

The power delivered per unit of terminal section of a nozzle is greater in the case of the gas turbine. If the gas escapes at a temperature of 700 degrees C. absolute, and at atmospheric pressure, the weight of a cubic metre is about 0.5 kilogramme. The discharge, by weight, will then be as follows:

for	$w_2 = 1000\ ^m/_s$	$1500\ ^m/_s$	$2000\ ^m/_s$
the values	1.8 kg.	2.7 kg.	3.6 kg.

per square millimetre of cross-section.

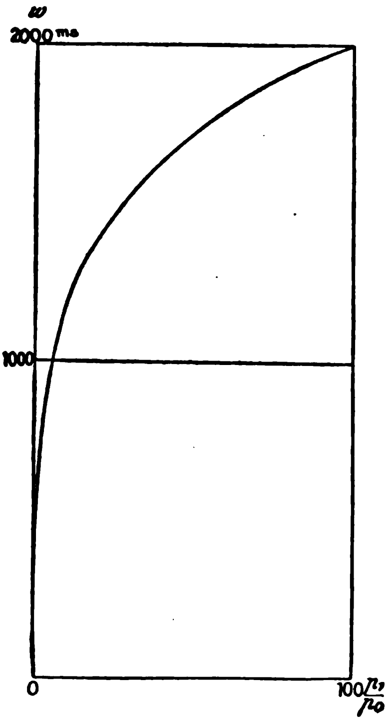


FIG. 65.—Velocity of discharge of steam from nozzles. Cycle with isobaric combustion; exhaust at 700 degrees.

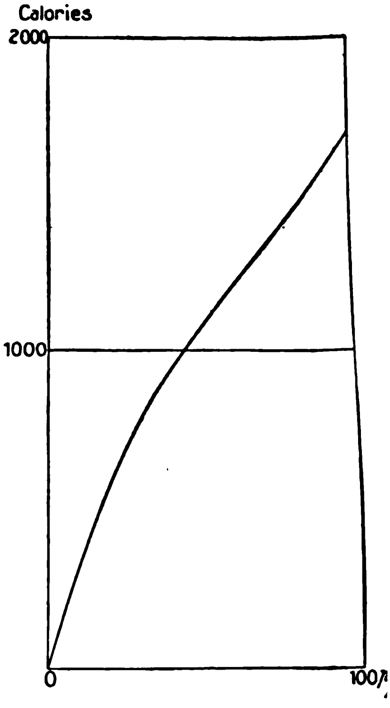


FIG. 66.—Energy in calories delivered per square millimetre of terminal cross-section of nozzle.

The power delivered in the form of kinetic energy per square millimetre of section will then be equivalent to the following quantities of heat, if we consider the same group of cycle as before:

Pressure of combustion.	5	10	20	30	40	60	80	100
Discharge in weight, kg..........	1.6	2.1	2.5	2.75	2.95	3.20	3.4	3.6
Power delivered, calories........	165	340	590	780	950	1240	1460	1700

If the exhaust takes place under a pressure reduced to $\frac{1}{n}$ atmosphere the figures for the power delivered will be reduced in the same proportion.

In a steam turbine operating with a pressure of 10 atmospheres and against a pressure of $\frac{1}{10}$ atmosphere in the condenser, the power delivered per square millimetre of nozzle is equivalent to only 60 calories.

The Formula of Saint Venant.

If a gas flows through a passage without friction, the initial and final pressures being ω_1 and ω_2, and the velocities w_1 and w_2, the adiabatic flow is represented by the formula of Saint Venant:

$$\frac{w_2^2-w_1^2}{2g}=\int_{\omega_2}^{\omega_1} v\,d\omega=\frac{\gamma}{\gamma-1}\omega_1 v_1\left[1-\left(\frac{\omega_2}{\omega_1}\right)^{\frac{\gamma-1}{\gamma}}\right] \tag{3}$$

or

$$\frac{w_2^2-w_1^2}{2g}=\frac{\gamma}{\gamma-1}(\omega_1 v_1-\omega_2 v_2). \tag{4}$$

Velocity of a Gas Flowing from a Reservoir through a Nozzle.

If we assume the velocity of approach w_1 as negligible, we have at any point of the nozzle corresponding to a pressure p_x, a velocity w_x given by the equation:

$$w_x=\sqrt{2g\frac{\gamma}{\gamma-1}\omega_1 v_1\left[1-\left(\frac{\omega_x}{\omega_1}\right)^{\frac{\gamma-1}{\gamma}}\right]} \tag{5}$$

and since

$$v_x=v_1\left(\frac{\omega_1}{\omega_x}\right)^{\frac{1}{\gamma}}$$

we have $$\Pi=\frac{s_x w_x}{v_x}=s_x\sqrt{\frac{2g\gamma}{\gamma-1}\frac{\omega_1}{v_1}\left[\left(\frac{\omega_x}{\omega_1}\right)^{\frac{2}{\gamma}}-\left(\frac{\omega_x}{\omega_1}\right)^{\frac{\gamma+1}{\gamma}}\right]}. \qquad (6)$$

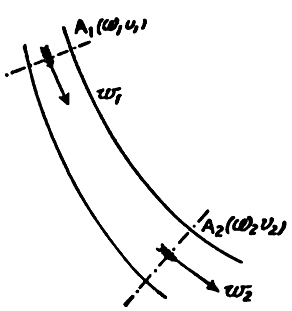


FIG. 67.—Diagram of nozzle velocities.

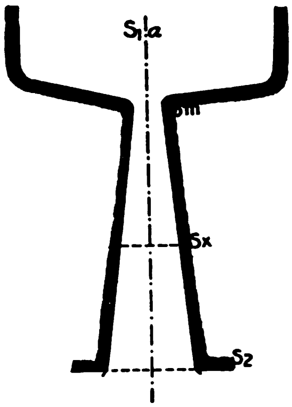


FIG. 68.—Diagram of nozzle sections.

We readily find that the minimum value of s_x (or of $\frac{v_x}{w_x}$) corresponds to a pressure ω_m, given by:

$$\frac{\omega_m}{\omega_1}=\left(\frac{2}{\gamma+1}\right)^{\frac{\gamma}{\gamma-1}} \qquad (7)$$

whence $$w_m=\sqrt{2g\frac{\gamma}{\gamma+1}\omega_1 v_1} \qquad (8)$$

and $$\Pi=s_m\sqrt{2g\frac{\gamma}{\gamma+1}\left(\frac{\omega_m}{\omega_1}\right)^{\frac{2}{\gamma}}\left(\frac{\omega_1}{v_1}\right)}. \qquad (9)$$

For air we have $\gamma=1.4$ and hence:

$$w_m=\left(\frac{2}{2.4}\right)^{3.5}, \quad \text{whence} \quad \omega_m=0.529\omega_1.$$

It can be demonstrated that p_m cannot fall below this value, and that w_m cannot exceed the velocity of sound.*

To release the air without loss of energy, down to the pressure of the atmosphere; or, more generally, down to any given pressure p_2, we must therefore use: if we have $\omega_2 > 0.529\omega_1$, a converging nozzle; if we have $\omega_2 < 0.529\omega_1$, a converging-diverging nozzle. The latter case is the only one to be considered for an air turbine, in which we always have $\omega_2 = 1$, and $\omega_1 > 1.9$ kilogrammes.

Length and Final Section of Nozzle.

In practice the diverging portion of the nozzle is made in the form of a cone of an angle of about 10 degrees, in order to avoid the breaking of the vein, which cannot follow the walls if a greater angle is used. The final section s_2, and hence the length, of the nozzle, will then be determined by

$$\Pi = \frac{s_2 w_2}{v_2} = \frac{s_m w_m}{v_m} \tag{10}$$

$$\frac{s_2}{s_m} = \frac{v_2}{v_m}\frac{w_m}{w_2} = \left(\frac{\omega_m}{\omega_2}\right)^{\frac{1}{\gamma}} \sqrt{\frac{1-\left(\frac{\omega_m}{\omega_1}\right)^{\frac{\gamma-1}{\gamma}}}{1-\left(\frac{\omega_2}{\omega_1}\right)^{\frac{\gamma-1}{\gamma}}}}. \tag{11}$$

Since, as for air, we have $\omega_m = 0.529\omega_1$, we have

$$\frac{s_2}{s_m} = \left(0.529\frac{\omega_1}{\omega_2}\right)^{\frac{1}{\gamma}} \sqrt{\frac{1-(0.529)^{\frac{\gamma-1}{\gamma}}}{1-\left(\frac{\omega_2}{\omega_1}\right)^{\frac{\gamma-1}{\gamma}}}}. \tag{12}$$

* The velocity of sound in a gas of which the absolute density is D is given by the formula of Newton:

$$v = \sqrt{\frac{E}{D}}$$

E, being the coefficient of elasticity of the gas, has for its value $\frac{C}{c} \times p$. We then have $V = \sqrt{\gamma p v}$.

Thus, for example, we have, for

$$\frac{\omega_1}{\omega_2}=5, \qquad 20;$$

$$\frac{s_2}{s_m}=1.35, \qquad 2.91.$$

The diagram (Fig. 69) shows these results. It will seen that the ratio of cross-sections for a given expansi is less in the case of air than for steam. The nozzle wi therefore, be shorter.

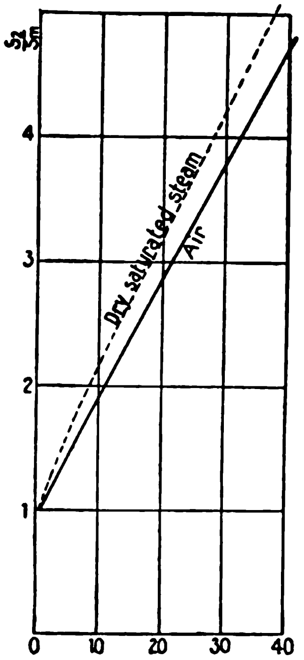


FIG. 69.—Ratio of nozzle sections for air and saturated steam.

This relation depends wholly on $\frac{\omega_1}{\omega_2}$ and not on the te perature.

Velocity in the Neck of the Nozzle.

The velocity w_m in the neck of the nozzle depends u the absolute temperature θ_1, and not upon the relation the pressures, as is shown in equation (8), in which we may replace $\omega_1 v_1$ by $R\theta_1$

$$w_m = \sqrt{2g \frac{\gamma}{\gamma+1} R\theta_1} \, . \tag{13}$$

Thus we have for:

$$T_1 = 1000 \quad 1500 \quad 2000 \quad 2500$$
$$\omega_m = 484 \quad 593 \quad 685 \quad 765.$$

metres per second.

Velocity of Discharge.

If, in equation (5), we note that:

$$\frac{\theta_1}{\theta_2} = \left(\frac{\omega_1}{\omega_2}\right)^{\frac{\gamma-1}{\gamma}}$$

we have:

$$w_2 = \sqrt{2g \frac{\gamma R}{\gamma - 1} \theta_2 \left[\left(\frac{\omega_1}{\omega_2}\right)^{\frac{\gamma-1}{\gamma}} - 1\right]} = \sqrt{2gCE(\theta_1 - \theta_2)} \tag{14}$$

which demonstrates the correctness of our original result based upon the principle of the conservation of energy.*

Influence of the Lower Pressure.

We have already seen that the velocity in the neck of the nozzle is entirely independent of the expansion ratio, and depends wholly upon the temperature of the gas before the expansion. Thus, in the case of air, the pressure in the neck of the nozzle is

$$0.529\omega_1.$$

Beyond the neck the expansion continues and the velocity increases regularly, while at the same time the pressure falls.

* Equations (3) to (14) have been taken from Stodola's treatise on the steam turbine; also figures 70 and 71.

If the cross-section increases as the square of the distance from the neck (a conical nozzle) the pressure varies according to a law which we may determine by taking the value of the velocity at each point (which is dependent upon the section), calculating the resulting variation in kinetic energy (from the neck to the point under consideration), and thence obtaining the temperature and the pressure for the given point, according to the law of adiabatic expansion.

If the angle of opening is given it will then be possible to determine a definite pressure for the terminal section as a function of the length of the nozzle.

The question arises: What will be the result if the medium into which the gas is discharged from the nozzle has a different pressure from that at the end of the nozzle?

The experiments of Professor Stodola upon steam have shown that if the pressure is lower than that of the exhaust it will produce sound waves, the pressure varying according to a sinusoidal curve in the discharge chamber.

Emden has calculated for air, and Prandtl for the vapor of water, the corresponding wave lengths.

The formula, of the form:

$$\lambda = a\left(\frac{w_m^2}{c^2} - 1\right)^{\frac{1}{\beta}}$$

in which c is the velocity of sound in the discharge chamber, and w_m the velocity of the fluid at its discharge, shows that the waves can be produced only when the velocity of discharge is greater than that of sound (Fig. 70). Professor Stodola admits that the fluid leaving the nozzle expands at once to the pressure of the surrounding medium, this causing the transformation of an excess of the potential energy of the exhaust fluid into living force. It is this excess which causes the sound waves and which is transformed into heat by friction and eddies.

If the pressure of the discharge chamber is greater than that corresponding to the terminal section of the discharge nozzle a sudden shock will be produced, causing a rebound in the pressure curve, followed by strong waves (Fig. 71).

This rebound may even force its way back into the nozzle, as shown in curve D.

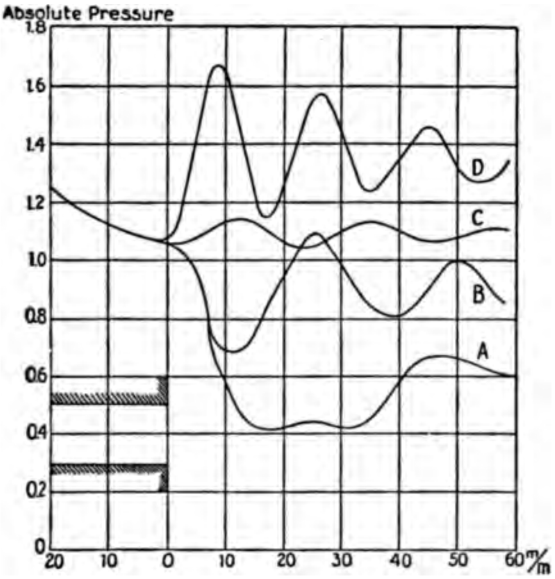


FIG. 70.—Pressure waves in discharge nozzles.

If the counter pressure be increased until it reaches the same order of magnitude as the original pressure, the rebound will penetrate further and further back into the nozzle, and may reach as far as the neck. This condition will be attended with a great loss of energy.

The theory of shock has been studied by Lord Rayleigh, Weber, Grashof, Lorenz, Prandtl and Proell, Stodola, and others. These researches, of much theoretical interest, lead to the following practical conclusions:

In order to secure adiabatic expansion from p_1T_1 to p_2T_2, it is necessary to use a nozzle of well-defined length. If too short the expansion will be incomplete, and the living force produced will not utilize all the available energy. If it is too long, a shock will be produced, accompanied by a loss of kinetic energy. The best length of nozzle can be determined only by experiment. In practice the pressure may be varied until the best result is produced. It will, therefore, be seen that it is difficult to obtain a good efficiency with an explosion turbine, in which p_1 and T_1 are continually varying, and in which the nozzle will be alternately either too short or too long.

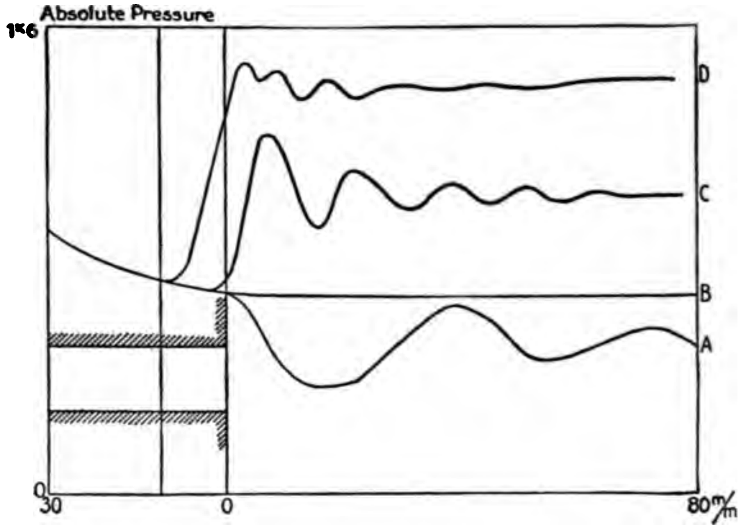


FIG. 71.—Oscillations in discharge nozzles.

It is also evident that any method of regulating a turbine by varying the maximum pressure must be accompanied by a loss of kinetic energy. It is possible that a compensating action may be obtained by varying the temperature and the maximum pressure. However this may be, the best method for regulating a turbine is by the variation of the admission.

Influence of Friction.

If, according to Stodola, we designate the successive states of the gas in two distinct sections of the fluid vein by the indices 1 and 2, and let Q_s be the quantity of heat dissipated by radiation and conductivity during the period in which the gas is passing from one of these sections to the other, we have:

$$(u_1+Ap_1v_1)-(u_2+Ap_2v_2)=AE+Q_e+A\left(\frac{w_2^2}{2g}-\frac{w_1^2}{2g}\right)$$

which gives the formula of Zeuner:

$$\lambda_1-\lambda_2=AE+Q_e+A\left(\frac{w_2^2}{2g}-\frac{w_1^2}{2g}\right).$$

From this, taking the particular case of adiabatic flow, we have:

$$\frac{w_2^2}{2g}-\frac{w_1^2}{2g}=\frac{1}{A}(\lambda_1-\lambda_2)$$

and since $\lambda=\text{constant}+CT$

$$\frac{w_2^2}{2g}-\frac{w_1^2}{2g}=\frac{C}{A}(T_1-T_2).$$

If, upon an infinitely small element of the gaseous mass we consider the influence of the walls to act in such a manner as to add or subtract a quantity of heat dQ, and, on the other hand, there is disengaged by friction a quantity of heat dR, we have:

$$dQ+dR=du+Apdv.$$

If we thus introduce the idea of frictional resistance, we find that the formula of Saint Venant thus generalized becomes, in the case of adiabatic flow ($E=0$, $Q_e=0$)·

$$\lambda_1-\lambda_2=A\left(\frac{w_2^2}{2g}-\frac{w_1^2}{2g}\right)=A\int_1^2 vdp-R.$$

As Professor Stodola has remarked, an adiabatic flow with friction is not adiabatic in the same sense as a flow without friction.

The work of friction is not wholly lost, for a portion is transformed into heat and acts to reheat the fluid, thus increasing the amount of heat transformed into work in the course of the expansion.

In the entropy diagram, the successive states of the fluid during the flow being represented by the curve A_1, A_2, the area $A_1\ A_2\ A_2''\ A_1''$ represents the total work of friction, (R), while the effective loss of kinetic energy is only the area $A_2'\ A_2\ A_2''\ A_1''$.

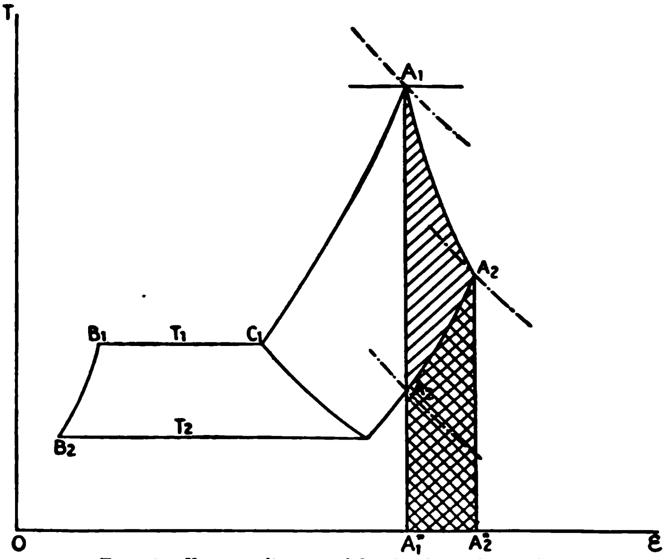


FIG. 72.—Entropy diagram of friction losses in nozzles.

Messrs. Delaporte, Lewicki, and Stodola have each given the figures resulting from their investigations with steam, showing the actual losses of kinetic energy in a diverging nozzle to be from 5 to 15 per cent.*

The actual loss of kinetic energy may be expressed by

$$Z = A\zeta \frac{l}{d} \frac{w^2}{2g}$$

in a cylindrical tube. For a conical nozzle it is necessary to proceed by integration.

* Delaporte: Nozzle of 6 to 9 mm. on 50 mm., discharging into the atmosphere, loss 5 per cent.

Lewicki: Nozzle of 6 to 7 mm. on 30 mm., ratio of pressures 6.86, loss 8 per cent.

Stodola: Nozzle of 12 mm. on 150 mm., losses 10 to 20 per cent.

Classification of Gas Turbines.—Efficiencies of Revolving Discs. Losses in the Blades.

The turbine may be either axial or radial, and composed of one or several discs.

The characteristic feature, however, in all cases, is the difference in pressure before and beyond the blades. If this difference is zero, that is to say, if the pressure in the space comprised between the guide blades and the revolving wheel is the same as at the discharge of the revolving buckets, we have an impulse turbine. In turbines of this class the transformation of energy takes place in the guide nozzles, and the revolving wheel utilizes the force thus developed.

If, on the contrary, a difference of pressure exists we have to deal with a reaction turbine.

The reaction turbine, having a degree of reaction of 1:2, operates with a velocity 40 per cent. greater than that necessary for an impulse turbine. For the same tangential velocity it therefore requires double the number of revolving wheels.

For these reasons, and especially because the impulse turbine lends itself to the use of a single expansion and does not require the gases to be discharged upon the revolving wheel until after they have been cooled by their expansion, the reaction type has not been applied to the gas turbine.

The impulse turbine with several wheels may be constructed either with velocity stages or with pressure stages.

The advantages of using multiple discs lies in the fact that the same hydraulic efficiency is obtained with one-third the tangential velocity if two revolving discs are used instead of one, and so on.

Unfortunately it is not yet possible to determine in advance the efficiency of the revolving discs of future gas turbines. We know what the hydraulic efficiency should be, neglecting the resistance of friction, but it is impossible, without previous experiments, to determine what the friction of the hot gases, more or less expanded, will be upon the fixed and moving blades.

It seems as if the friction should be less than in the case of the steam turbine. If so, the number of revolving discs might be increased, thus increasing both the total efficiency and the hydraulic efficiency, properly so called.

In steam turbines of the impulse type the present tendency is to multiply the number of pressure stages and to reduce the number of revolving discs in each stage. It would not be easy to carry this method very far with the gas turbine since it would be necessary to have a final temperature of less than 700 degrees absolute, in order to avoid having too high temperatures in the upper pressure stages. This, as we have already seen, would necessarily be accompanied by a reduction in the thermal efficiency.

There are, therefore, in the case of the gas turbine, certain contradictory influences, of which the final effect upon the mechanical efficiency can be determined only by experiment. Nevertheless, everything indicates that the mechanical efficiency of the gas turbine should be at least equal to that of the steam turbine.

Friction of Discs Revolving in Air.

In existing steam turbines, the friction of the revolving discs is the cause of a certain loss of energy. This loss is less than was formerly believed, but it is nevertheless of importance. We shall see that this friction loss will be relatively less in the gas turbine, even when the pressure of the fluid is equal to that of the atmosphere. At the present time we have no means of knowing the magnitude of this resistance when the machine is operating under load. The investigations of Odell, Lewicki, Stodola, and others, have enabled us to determine the amount of power absorbed by friction when operating without load.

It should be noted that this power is from two to four times greater for a bladed disc revolving in the atmosphere than when it is enclosed in a chamber, sealed against all ventilation. Under such conditions, a bladed disc absorbs but little more work than a flat disc. The work of friction is proportional to the cube of the number of revolutions and may be represented by

$$N_0 = aD^2\left(\frac{u}{100}\right)^3\delta$$

in which a is a coefficient, D the diameter of the disc, u the tangential velocity, and δ the specific gravity of the gas. All other things being equal. there is absorbed in saturated steam 1.3 times the amount of work that is absorbed in air, a point which is favorable to the gas turbine.

Under atmospheric pressure, steam superheated to 300 degrees Centigrade offers the same frictional resistance as air at the ordinary temperature.

Lewicki has found that at 300 degrees, and under a *very low pressure*, the amount of work absorbed is a little less than in air *at atmospheric pressure.*

In the case of a gas turbine, however, with an exhaust of 600 degrees absolute, the density of the air is only

$$1 \times \frac{273}{600} = 0.46$$

and we conclude that the loss would probably be less than that observed in the steam turbine.

There is another factor which should reduce still further the relative importance of this loss. This is the relatively small size of the discs of the gas turbine. We have already seen that the total section required for the passage of the gas, for the same amount of work, is less than in the case of the condensing steam engine.

In consequence it may be possible, at least in the case of very large turbines, to reduce the diameter of the revolving wheels. For small units the diameter is controlled by the tangential velocity and by the necessary number of revolutions, and this may not be reduced. On the contrary, the higher velocity of the flow of air may lead to larger wheel diameters.

Regulation of Gas Turbines.

The question of speed regulation, a matter of great industrial importance, does not yet appear to have been made the object of any important effort. Let us see how the regulation of a turbine with a single nozzle may be effected. There are two cases to be considered, that in which the compressor is not mechanically connected to the turbine wheel, and that in which a direct mechanical or electrical connection exists between these two elements.

In the first case: The relation

$$\frac{w_2^2}{2g}=\frac{\gamma}{\gamma-1}R\theta_1\left[1-\left(\frac{\omega_2}{\omega_1}\right)^{\frac{\gamma-1}{\gamma}}\right]=\frac{\gamma}{\gamma-1}R\theta_2\left[\left(\frac{\omega_1}{\omega_2}\right)^{\frac{\gamma-1}{\gamma}}-1\right]$$

shows that the power delivered by a nozzle is not influenced very strongly by a variation in the temperature before expansion θ_1 (which is the temperature of combustion), if the ratio of pressures is not varied.

In fact the kinetic energy of a unit of mass is proportional to θ_1 but the density of the gas at discharge is proportional to $\frac{1}{\theta_1}$; the pressure ω_2 not varying. The velocity w_2 being proportional to $\sqrt{\theta_1}$ the power developed, proportional to $\frac{mw_2^2}{2}$, is proportional to $\sqrt{\frac{\theta_1}{\theta_1}} \times \theta_1$, or to $\sqrt{\theta_1}$.

If we vary the ratio of pressures $\frac{\omega_1}{\omega_2}$ without modifying θ_1 we obtain a variation of θ_2 proportional to $\left(\frac{\omega_2}{\omega_1}\right)^{\frac{\gamma-1}{\gamma}}$. From this the density varies as $\frac{1}{\theta_2}$, in supposing that ω_2 is fixed and that ω_1 alone varies. The flow in weight is then proportional to

$$\frac{1}{\theta_2}\sqrt{1-\left(\frac{\omega_2}{\omega_1}\right)^{\frac{\gamma-1}{\gamma}}}$$

and the product $\frac{mw_2^2}{2}$ is proportional to

$$\frac{\sqrt{1-\left(\frac{\omega_2}{\omega_1}\right)^{\frac{\gamma-1}{\gamma}}}}{\left(\frac{\omega_2}{\omega_1}\right)^{\frac{\gamma-1}{\gamma}}}\left[1-\left(\frac{\omega_2}{\omega_1}\right)^{\frac{\gamma-1}{\gamma}}\right].$$

If, therefore, we act solely on the *temperature of combustion,* we may reduce the power in the ratio of $\sqrt{\theta_1}$, and the temperature of the exhaust will be reduced when operating at light loads. The efficiency will not be materially affected (it will be rigorously unchanged if the compression is adiabatic).

If we act solely on the *pressure of combustion* the temperature of the exhaust will be increased as the power is reduced, a condition which is entirely inadmissible, since it would lead to higher exhaust temperatures than 700 degrees at light loads. If, on the contrary, we arrange that the exhaust temperature shall not exceed 700 degrees at light loads, we shall have a lower exhaust temperature under full load, and the efficiency will be reduced.

This latter inconvenience may be avoided and a more energetic regulation effected by causing *the temperature and the pressure to vary at the same time,* in such a manner as to keep the final temperature θ_2 constant, although this method will lower the thermal efficiency at light loads.

Under these conditions the density does not vary; the velocity varies as

$$\sqrt{\left(\frac{\omega_1}{\omega_2}\right)^{\frac{\gamma-1}{\gamma}}-1}$$

and the energy developed varies as

$$\sqrt{\left(\frac{\omega_1}{\omega_2}\right)^{\frac{\gamma-1}{\gamma}}-1}\times\left[\left(\frac{\omega_1}{\omega_2}\right)^{\frac{\gamma-1}{\gamma}}-1\right].$$

If the reduction in pressure is produced by a valve placed between the combustion chamber and the compressor, the efficiency will be reduced to an inadmissible extent. In such a case it will be advisable to use an independent compressor, with variable compression.

The variations in flow resulting from variations of θ_1 or of $\frac{\omega_1}{\omega_2}$ or from both of these elements at the same time, will cause the nozzles to operate under different conditions from those for which they were designed, when the machine is under light loads. This will reduce their hydraulic efficiency; on the contrary, the efficiency of the revolving discs will be increased. Another method applicable to the single nozzle is based on the hit-or-miss system. So, far as efficiency is concerned, this is probably the best method, since the volume of the combustion chamber is so small compared to the flow of gas that the transition periods of low hydraulic efficiency would be of negligible duration. On the other hand, some method of re-ignition would be required, unless the compression and temperature were sufficient to insure automatic ignition.

If the compressor is mechanically or electrically connected to the turbine, it may be included in the regulation system by causing the excess production to be accumulated in a reservoir during the periods when the demand for the turbine is below normal.

Let us now consider turbines with multiple nozzles. In such cases it will be necessary to determine, according to the conditions of each particular machine, whether it is better to modify the discharge of a single nozzle or to act proportionally upon them all.

The true solution appears to be that employed in the Curtis steam turbine, by shutting off a certain number of nozzles without affecting the others. This method should be accompanied by the use of a fly-wheel (or accumulator of energy) if great regularity of speed is required, or if there are not many nozzles.

It will be necessary to control the gas and combustible at each nozzle by two valves. The regulator will act, through an auxiliary motor, directly upon the air valves, and upon the valves controlling the combustible by means of a small piston, acting only when the pressure in the combustion chamber reaches a certain value; its movement being, to a certain extent, proportional to this pressure.

However all these points may be, it is evident that the question of the regulation of the gas turbine should not present any serious difficulties.

Details of Gas-Turbine Construction.

Air Compressors.

We have seen that it is necessary to have as high a ratio of pressures as possible in order to attain a high efficiency, both thermal and mechanical.

It appears that the gases which are delivered into the combustion chamber should, therefore, be compressed to about 40 atmospheres; or else that a lower pressure, say about 10 atmospheres, be used in connection with the maintenance of a pressure of about $\frac{1}{4}$ atmosphere in the exhaust space. In the second case it is evident that we should have to use a second compressor to raise the pressure of the exhaust gases from $\frac{1}{4}$ atmosphere up to atmospheric pressure.

In order that the work required for this operation should not exceed that required to produce the higher initial pressure of 40 atmospheres it is necessary that the absolute temperature of the exhaust gases should be kept down to a minimum value T_0 which corresponds to the temperature of the atmosphere. This is a difficult matter, and can be effected only by using regenerators or tubular coolers of large size; or by using injections of water, involving the employment of a wet air pump, or possibly a barometric condenser and dry air pump.

With these reservations the two systems may be considered as equivalent, but that which involves the employment of an air pump is principally of interest so far as it bears upon the practicability of replacing reciprocating compressors with turbine air compressors.

In practice it appears to be difficult to attain higher degrees of compression than 20 to 25 atmospheres with the turbine compressor. To produce such pressures it is necessary to have a large number of revolving wheels, rotating at high velocities; and as these must be in a fairly dense medium the mechanical losses resulting from the friction of the wheels in the compressed air must have an injurious influence upon the efficiency of the machine.

If it is desired to avoid the use of piston compressors it appears to be necessary to subdivide the operation in the manner indicated hereafter. If, on the contrary, compressors of the reciprocating type are permissible, there is no difficulty in attaining pressures as high as 50 atmospheres. These machines, as we shall see, have an excellent efficiency, which is a most important feature in the present instance. At the same time they add greatly to the bulk of the apparatus, and take from the gas turbine many of the advantages possessed by the steam turbine.

In cases in which the bulk of a large compressor is a matter of importance it may be found desirable to have the reciprocatory compressor preceded by a turbine compressor, or by an ordinary rotary compressor.

For example, we may use a turbine compressor to compress the air to a pressure of 4 atmospheres, which involves an amount of work equivalent to only 30 calories per kilogramme, and then complete the compression by the use of a piston compressor, which would raise the pressure from 4 to 40 kilogrammes per square centimetre (4 to 40 atmospheres) with an expenditure equal to 46 calories.

The mechanical efficiency of the turbine compressor being assumed to be equal to 0.70 and that of the reciprocating compressor to 0.85, the combination would have an efficiency of 0.59, but the bulk of the piston compressor would be reduced to one-fourth that otherwise necessary, since the volume of air to be handled is reduced to one-fourth.

We then have as available the following three plans:

A piston compressor, compressing the air directly to 40 atmospheres; a turbine compressor, compressing the air to 4 atmospheres, followed by a piston compressor raising it to 40 atmospheres; or a turbine compressor, compressing the air to 10 atmospheres, and a cooler beyond the gas turbine, followed by a turbine air pump taking the burned gases at $\frac{1}{4}$ atmosphere and discharging them into the atmosphere.

Ordinary Piston Compressors.

An excellent study of machines of this type was given by M. Barbet in the *Génie Civil* from May to August, 1895. We may here note at once that compressors with liquid pistons, such as those of Sommellier, Dubois, and François, and others, are too bulky to be considered in connection with the gas turbine, and we are, therefore, obliged to resort to compressors of the Colladon type, with simple injection of water; or of the Mekarski type, using several stages of compressors, with intercoolers.

M. Mekarski has constructed a number of vertical compressors, operating at 100 to 150 revolutions and compressing air to about 60 atmospheres, taking 0.065 kilogramme of air per revolution. These machines have four single-acting cylinders, arranged in tandem pairs, having automatic valves, and provided with water injection in the low pressure cylinders, the high-pressure cylinders being jacketed. The volume efficiency was about 67 per cent. The efficiency in work may be computed as follows: The indicated power of the engine operating the compressor was 97 horse-power, and there was compressed 390 kilogrammes of air per hour.

Theoretically, an isothermal compression to 60 atmospheres would require:

$$\frac{42,308 \times 390}{270,000} = 61.1 \text{ h.p.}$$

The efficiency of the combined air compressor and steam engine, was, therefore, $\frac{61.1}{97} = 0.63$.

If we assume the efficiencies of the compressor and the steam engine to be the same, their value would be $\sqrt{0.63} = 0.79$. This shows that even with a piston compressor of such a moderate size as 100 horse-power a mechanical efficiency of nearly 0.8 may be obtained. Under these conditions one horse-power delivered to the shaft of the compressor would furnish 5 kilogrammes of air compressed to 60 atmospheres.

Taking compressors of large size we may cite the vertical triple expansion machines of 2000 horse-power, built by the Creusot Works for the Paris Compressed Air Company. The air, in these machines, is raised to 7 atmospheres, the ratio thus being 1:7. The air cylinders are double acting, and are arranged tandem with the steam cylinders. The compression is effected in two stages, with a pressure of 2.7 kilogrammes per square centimetre (38.4 pounds per square inch) in the intermediate reservoir. The air cylinders have no water jackets, but water injection is used in the cylinders, in the valve chests, and in the intermediate reservoir.

Operating at 50 revolutions per minute, these machines handle 20,720 cubic metres of air, or 25,400 kilogrammes per hour, with a volumetric efficiency of about 0.80.

According to tests conducted by M. Bourdon and by Professor Gutermuth, the expenditure of one horse-power indicated on the pistons of the steam engine compresses 12.7 kilogrammes of air to the ratio of 1:7. Other tests give a combined efficiency for the engines and compressors of about 0.90.

In these machines of the Paris Compressed Air Company the curve of compression corresponds closely to the equation

$$pv^{1.3} = \text{constant}$$

and the air enters at a temperature of +5° C. and leaves at +25°, while an adiabatic compression would give a final temperature of +200°. The cooling by water injection thus appears to be very effective.

As a final example of a compressor operating at slow speed we may cite a Strnad machine, of which tests by Professor Gutermuth upon one of 300 horse-power gave the following results:

Ratio of compression	1 to 7.1
Number of revolutions	50 to 70

$$\text{Ratio: } \frac{\text{theoretical work of compression}}{\text{indicated power of engine}} = 0.73$$

$$\text{Ratio: } \frac{\text{indicated work of compression}}{\text{indicated work of engine}} = 0.84.$$

It follows that the mechanical efficiency of the compressor, assuming it equal to that of the engine would be $\sqrt{0.84} = 0.92$, but since the elevation of temperature has the effect of raising the value of the ratio

$$\frac{\text{theoretical work of compression}}{\text{indicated work of compression}}$$

to a value 0.87, we have in reality an efficiency $0.87 \times 0.92 = 0.80$. The Strnad compressor is a two-stage machine with two cylinders in tandem with the steam cylinders of a compound engine. It is characterized by an injection of water under pressure in the cylinders, and by the use of Corliss valves. The consumption of water is 2.55 litres per cubic metre of air. With this machine one horse-power compresses 12 kilogrammes of air to a ratio of 1:7.

The air compressors at the Billancourt Works of the General Omnibus Company of Paris have an indicated power of 875 horse-power, and compress 3420 kilogrammes of air per minute to a pressure of 80 kilogrammes per square centimetre (1137.6 pounds per square inch), making 32 revolutions per minute. The compression is effected in three stages: 4 kg., 24.5 kg., and 80 kg.

There are injected two kilogrammes of water for every kilogramme of air in the low-pressure cylinders. With the external air at 18° C. (64.4 F.) the temperature of the compressed air is 60° C. (140 F.).

We have:

$$\frac{\mathfrak{T}_c}{\mathfrak{T}_i \text{ of the steam engines}} = 0.53 \text{ to } 0.55$$

Whence the efficiency is 0.73 to 0.74.

We see, therefore, that the older, slow-running compressors which we have examined give values which may be taken as:

0.70 to 0.80 for machines of 100 to 1000 h.p.

0.80 to 0.90 for units of 2000 h.p.

Modern High-Speed Compressors.

During the past few years the necessity for providing machines adapted for direct connection to electric motors has led designers to produce high-speed air compressors, capable of being operated at speeds of 500 to 600 revolutions per minute for sizes of 100 horse-power.

At the Düsseldorf exposition there was shown by Messrs. Pokorny and Wittekind, of Frankfort, a compound tandem compressor with Koster valve gear, driven by an electric motor of 70 horse-power, running at 550 revolutions.

The stroke of pistons was 150 mm. and the diameters of cylinders 300 and 190 mm. respectively. This machine compressed about 750 kilogrammes of air per hour to a pressure of 7 atmospheres. Data concerning tests upon modern high speed compressors will be found in the papers of Richter, Lebrecht, Biel, etc.*

Machines of this type offer advantages so far as reduction in dimensions are concerned, especially if preceded by a turbine compressor. If the latter machine is used to compress the air to a pressure of 4 atmospheres absolute, the small machine described above would be available to compress 3000 kilogrammes of air from 4 to 28 atmospheres. This would be sufficient for a gas turbine of 600 horsepower.

Rotary Compressors.

Rotary compressors have not, up to the present time, been very successful. They are, however, used to some extent for moderate and small powers.

The *Compagnie Générale Electrique* of Nancy has developed the Hult rotary steam engine, the principle of which is applicable to the compression of air.

Messrs. Siemens and Halske have produced a rotary compressor, adapted only to small sizes, and, being without clearance space and capable of direct connection to electric motors, may be used for compressions of a ratio 1:3, or to produce a vacuum as low as 1.5 mm. of mercury. It would be interesting to know the efficiency of these machines, as they might be applicable for use with an initial compression of 4 atmospheres and an exhaust pressure of $\frac{1}{10}$ atmosphere, to give an expansion ratio of 1:40. These machines would be adapted only to small gas turbines.

* Richter: Thermische Untersuchung an Kompressoren, Zeitschr. d Ver. deutscher Ing., July, 1905. Lebrecht: Versuche mit raschlaufende gas-compressoren, Zeitschr. d. Ver. deutscher Ing., 1905. Biel: Zeitschr. d. Ver. deutscher Ing., 1905, p. 540.

Turbine Compressors.

In 1902 M. Rateau directed attention to the properties of high-speed blowers, showing such a machine connected directly to a steam turbine making 10,000 to 20,000 revolutions per minute. This blower, with an efficiency of about 0.60, gave a compression ratio of 1 to 1.5. By coupling a series of such blowers the pressure may be increased in a geometric proportion, giving with four such blowers a pressure of $(1.5)^4$, or about 5 atmospheres absolute.

Theoretically, the efficiency of such an arrangement is equal to the efficiency of a single wheel. Nevertheless, it does not appear to be practicable to use this machine for pressures much higher than 5 atmospheres, since the frictional resistance in atmospheres of higher densities will increase to an extent which causes the efficiency to fall off materially.

If a machine of this type is used to lower the exhaust pressure of a turbine, the above inconvenience does not appear, and a greater number of wheels will not be required to produce the desired reduction in pressure. At the same time the apparatus will have to be much larger for the same discharge of gases by weight.

For example, if it is desired to produce a pressure of $\frac{1}{5}$ atmosphere, the first wheel will produce a reduction of only $\frac{0.5}{5} = 0.1$ atmosphere.

It follows that n wheels will give a final pressure equal to $0.2 \times (1.5)^n$, and we should have $0.2 \times (1.5)^n = 1$, whence $(1.5)^n = 5$, and $n = 4$. It is understood that this assumes that the exhaust gases have been cooled to the temperature of the air before entering the apparatus.

If this arrangement is employed it is very desirable that the multicellular blower should be connected directly to the gas turbine; but since the tangential velocity of the blower should reach 260 metres per second to obtain the pressure ratios given above, a high rotative speed becomes necessary (20,000 revolutions in a machine of 200 horse-power compressing air from 1 to 5 atmospheres).

If the gas turbine is designed for driving dynamos, etc., it should make not more than 1000 to 2000 revolutions. In such cases it would probably be advisable to have a separate turbine to drive the blowers.

Besides the investigations of M. Rateau several other attempts have been made to utilize the turbine compressor.

In England experiments have been made with the Parsons turbine, while similar attempts have been made by the General Electric Company to utilize the Curtis turbine. No practical results, however, have yet been made public.

The idea of Burdin and Tournaire has, therefore, not yet entered into practical operation, but it appears to be on the eve of successful application.

Thermal Regenerators.

The application of alternating regenerators, necessary for use with piston engines, does not appear to have given practical results. The question appears in a different light, however, when considered in connection with the gas turbine.

In the latter case the gases issue from the turbine in a continuous manner, with a fairly high velocity, which is a great advantage in connection with the transmission of heat.

The gases to be heated are also delivered in a continuous manner by the compressor, and in view of the high pressures under consideration the losses involved in producing a rapid circulation in a system of tubes need not be very serious.

The burned gases discharged by a gas turbine would be much cleaner than those dealt with in the case of superheaters attached to steam boilers, or than those discharged from reciprocating gas engines; the continuous combustion under pressure being that which leaves a minimum amount of residue. We have also seen that these gases have a very slight oxidizing action. These conditions are all favorable to the use of the regenerator.

Regeneration from Gas to Gas.

Devices intended to heat the compressed air by means of the heat abstracted from the exhaust gases, are analogous to steam superheaters and are operated under similar temperature conditions. They are less subject to oxidation, the gases being poor in oxygen, and the temperatures are more uniform.

The nature of the metal used is immaterial, so far as the transmission of heat from gas to gas is concerned. Tubes either of iron or steel may be employed. Copper or aluminum tubes are of interest only as regards increased durability. High circulation velocities are essential, and the counter-current principle should be adopted, taking the necessary structural precautions against injury from expansion and contraction.

Regeneration by Steam.

When the regeneration of the waste heat is effected by the aid of steam, the latter is generated in a boiler of the instantaneous flash type, heated by the exhaust gases. Such a boiler includes a reheater, followed by the boiler proper, and by a superheater, all with a systematic circulation. These three portions, however, are not necessarily clearly defined. We may use, for example, a boiler of the Serpollet type, or one of the system proposed by Col. Renard.*

The water, entering at one end, travels continuously through until it reaches the other extremity in the state of superheated steam. Here, as before, the gases should circulate very rapidly.

* See Génie Civil, 1905.

Production of the Combustible Mixture.

The combustible employed may be solid, liquid, or gaseous, but it is improbable that a turbine employing a solid combustible will soon be realized. It would undoubtedly be very difficult to inject a solid combustible into a combustion chamber against a pressure of 30 to 40 atmospheres, and the ashes discharged with the burned gases would cause excessive wear upon the blades of the turbine.

The gas turbine is, therefore, not adapted for the direct use of solid combustibles, and, for a long time, at least, the use of a gas producer must be considered necessary.

Gaseous Fuel.

As we have already seen, the lean gases are almost as advantageous as the richer gases. In the case of Dowson gas the air compressor must deliver a volume of air equal to 1.2 times that of the gas. This involves the use of two compressors of different power and capacity.

It may be suggested that the Gardie system is available for this purpose, in view of the fact that it utilizes the sensible heat of the gases which leaves the producer at about 700 degrees and which is ordinarily lost; but the operation of a gas producer under a pressure of more than 10 atmospheres appears to be a difficult problem. It would thus be necessary to operate the producer under a moderate pressure, a method practicable only when the turbine is operated with a low-pressure exhaust. The principal advantage of this system appears to lie in the avoidance of any wasting of the gas.

If we use the waste gases from blast furnaces it will also be necessary to have two compressors of different dimensions, the volumes being in the proportion of 0.7 to 0.9 of air to 1 of gas.

It would be difficult to use a regenerator in this case, for it is well known that the large amount of dust contained in furnace gases renders their use difficult under steam boilers. This difficulty, however, is not impossible of removal, and the use of the regenerator would enable improved efficiency to be secured.

With very rich gases, such as illuminating gas or acetylene, a very small compressor would be required for the gas. The relations of the volumes to be compressed would be, respectively, as 1 to 8 and as 1 to 20. The total value of the work of compression, however, is not much less than in the case of lean gases.

With liquid combustibles a very small pump, consuming but little power, is required, but the air compressor demands about as much power as the two pumps together in the case of gaseous fuel.

Liquid combustibles are used, either by volatilization in a carburetor, similar to those employed for gasoline motors, or by direct injection into the combustion chamber in connection with some sort of pulverizer or atomizer. It may also be found advantageous to heat or volatilize the liquid fuel by means of the heat of the exhaust gases.

In turbines using an injection of water or steam, the injection may be made either in the combustion chamber or before entering it. We are inclined to favor the latter solution, especially in the case of steam.

The Combustion Chamber.

The dimensions of the combustion chamber should be such that the reaction may be completed before the gases leaving the zone of combustion penetrate the nozzle.

According as the maximum temperature T_1 varies from 1000 degrees to 2500 degrees absolute, the velocity in the throat of the nozzle varies from 500 to 800 metres per second. If we take a temperature of 2000° C. absolute, we have a velocity of 685 metres per second. For the terminal part of the section of the nozzle we have, for a compression ratio of 25 to 30, and a temperature of 700° at the end of the expansion, a velocity of 1500 metres per second. What, then, should be the length of the combustion chamber, under these conditions? Taking a temperature of 2000° absolute, and a pressure of 25 atmospheres, the density will be: (25 to 30) $\times \frac{700}{2000} =$ 8.75 to 10.5 times that of the exhaust gases; let us say 10 times. If, then, the section of the combustion chamber is equal to that of the lower terminal section of the nozzle we should have a velocity one-tenth as great, or about 150 metres per second.

It is generally admitted that the velocity of propagation of flame, at atmospheric pressure, is only about 1 to 2 metres per second. It would therefore be necessary to give the combustion chamber a section of about 100 times that of the end of the nozzle, in order that the combustion may be absolutely complete. The best dimensions can be determined only by experience. It is hardly probable that such a size as indicated above would be necessary, for with the temperatures and pressures under consideration the combustible would ignite spontaneously, and hence the velocity of propagation of the flame would become almost infinite.

In practice it seems as if a cross-section 10 times as great as that of the terminal section of the nozzle ought to be sufficient. This would give a velocity in the chamber of about $\frac{1}{40}$ that of the discharge, or about 38 metres per second This velocity would be considered normal in ordinary gas mains, operating under atmospheric pressure.

The length of the combustion chamber may be fixed at 5 to 10 times its diameter, but there is no special rule governing this dimension.

For example, suppose we take a nozzle having a terminal section equivalent to that of a circle 10 mm. in diameter, and make the diameter of the combustion chamber 50 mm., we get a velocity in the latter of:

$$\frac{38 \text{ m. per sec.}}{2.5} = 15.2 \text{ m. per sec.}$$

If we give the chamber a length of 300 mm., the gas will be in it for about:

$$\frac{0.30 \text{ m.}}{15.2 \text{ m. per sec.}} = 0.02 \text{ second.}$$

This corresponds to the duration of the combustion in a gasoline motor making 1500 revolutions per minute.

Nozzles for Injection and Expansion.

In the case under consideration there are many reasons in favor of partial injection.

The injection nozzles, or ring of fixed blades, being subjected to the high temperature of combustion, it is necessary in their construction to provide a material possessing special properties, including resistance to heat and to chemical reactions, also to repeated expansion and contraction, and finally possessing considerable mechanical strength.

These conditions are well filled by the material carborundum, with the exception of the fact that precautions must be taken with respect to its rather high conductivity for heat.

The Combustion Chamber and the Nozzles for the Characteristic Portion of the Gas Turbine.

The simplest arrangement, applicable to small turbines, consists of the combination of a combustion chamber and single diverging nozzle, the latter preferably of rectangular cross-section. The terminal portion of the nozzle may be made of metal, since the gases at this point will be cooled to below 800°. The end of the nozzle may be divided by thin partitions, thus forming a portion of the distributing ring.

Construction of the Revolving Wheels.

The construction of the revolving wheels of the gas turbine does not differ materially from that already followed for the steam turbine. The arrangement of a single disc may be adopted or that of multiple discs; the latter is preferable, since the velocity of the fluid will be at least as high as in the steam turbine.

The practical design depends upon the linear speed, and the moderate velocity of 120 to 200 metres per second at the perimeter of the wheels may be maintained.

It must be considered that the temperature of 300° to 400° Centigrade is somewhat higher than ordinarily exists in the steam turbine, and it is desirable to select materials of which the strength is reduced as little as possible at these temperatures. No serious difficulties need be apprehended on this score. Steam turbines of the Laval type have been operated with steam superheated to 600° absolute and with air heated to 700° absolute, without any difficulty.*

Tests upon nickel steel have given the following results:

Absolute temperature.	300	500	600	700
Breaking load (kg. per cm.^{2})	81	91	92	73
Elastic limit (kg. per cm.^{2})	70	60	54	40
Elongation, per cent	10.7	8.7	8.3	7.0
Reduction of section, per cent	60.8	60	60.8	70

We may expect that the light metal blades, when subjected to the oxidizing action of the gases at these temperatures will wear somewhat more rapidly than the blades of the steam turbine. In the latter machine, however, it is admitted that the principal cause of wear is due to the action of particles of moisture. In the case of superheated steam, and consequently with the gases of the gas turbine, the wear

* See Lewicki: Zeitschr. d. Ver. deutscher Ing., 1905.

of the blades is negligible. The burned gases contain but a small proportion of free oxygen (5 to 10 per cent.), and nickel steel is especially resistant to oxidizing influence.

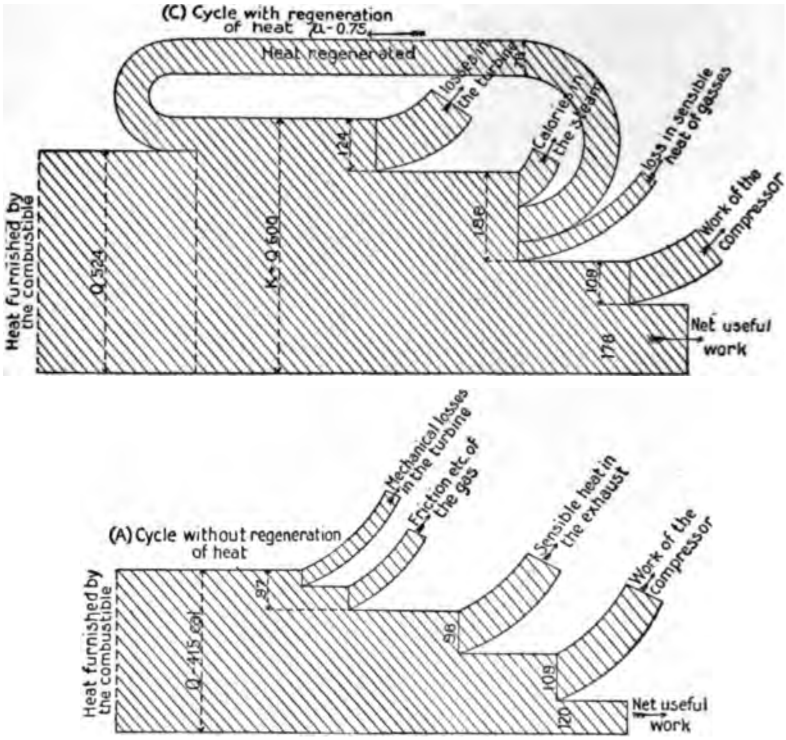


FIG. 73.—Diagrams showing the distribution of heat from the combustion of 1 kilogramme of gas. Water-injection not included. Isothermal compression; isobaric combustion at a pressure of 40 atmospheres.

General Design of a Gas Turbine.

Let us now apply the preceding theoretical calculations to three practical examples.

For this purpose we will select a system employing isothermal compression, with combustion under constant pressure; the turbine being intended to develop 150 horse-power, and thus capable of driving a dynamo of 100 kilowatts.

Preliminary Computations for a Gas Turbine of 150 H. P.

	A	B	C
Weight of gas consumed per hour P	5.5 kg. × 150 = 825 kg.	3.5 kg. × 150 = 525 kg.	5.5 × 150 = 825 kg.
Consumption of oil of 10,000 calories $\frac{PQ}{10,000}$	$\frac{825 \times 415}{10,000} = 34$ kg.	$\frac{525 \times 524}{10,000} = 27.2$ kg.	$\frac{825 \times 346}{10,000} = 28.4$ kg.
Consumption of oil per *effective* h. p. hour	0.233 kg.	0.182 kg.	0.188 kg.
Power absorbed at shaft by compressor $150 \frac{Nc}{Nu}$	150 × 0.95 = 143 h. p.	150 × 0.60 = 90 h. p.	150 × 9.95 = 143 h. p.
Effective power of the turbine $N = Nu + Nc$	150 + 143 = 293 h. p.	150 + 90 = 240 h. p.	150 + 143 = 293 h. p.
Absolute temperature of combustion	2050 degrees C.	2050 degrees C.	2050 degrees C.
Air of combustion per kilogramme of combustible	23 kg.	19 kg.	29 kg.
Minimum admissible for assumed ratio	15 kg.	15 kg.	15 kg.
Volume of exhaust gases at 700° absolute, and at atmospheric pressure	1600 cu. m. per hour	1200 cu. m. per hour	1600 cu. m. per hour
Same volume if exhausting at pressure of one-fifth atmosphere	8000 cu. m. per hour	6000 cu. m. per hour	8000 cu. m. per hour
Theoretical velocity of gases discharged from nozzle	1600 m. per sec.	1840 m. per sec.	1600 m. per sec.
Theoretical cross-section of nozzle at terminal	275 sq. mm.	168 sq. mm.	275 sq. mm.
Theoretical section with exhaust at one-fifth atmos	1375 sq. mm.	840 sq. mm.	1375 sq. mm.
Absolute temperature at throat of nozzle	1450 degrees C.	1450 degrees C.	1450 degrees C.
Volume of air aspired by the compressor at 300° absolute, and atmospheric pressure	620 cu. m.	400 cu. m.	620 cu. m.
Volume of low pressure cylinder of compressor at 120 revolutions	52 cu. dm.	34 cu. dm.	52 cu. dm.
Volume of cylinder of a 4-cycle gas engine of 150 h. p., at 115 revolutions	280 cu. dm.	280 cu. dm.	280 cu. dm.

The pressure ratio will be taken at 40, and this may be attained either with the exhaust at atmospheric pressure or with exhaust at $\frac{1}{4}$ atmosphere; in this latter case the pressure of combustion will be 9 atmospheres above perfect vacuum.

We will consider the following cases:

A. Illuminating oil with a lower calorific value of 10,000 calories, without regeneration.

B. Same data, with steam regeneration, with $\mu = 0.75$.

C. Same data, with regeneration of gas to gas, $\mu = 0.75$.

The results of the computations are given in the preceding table. The figures must be understood as having only a relative value, owing to the numerous hypotheses which have been made in deriving them. Their comparative value, however, remains unaffected. We see, for example, that the injection of steam permits the reduction in capacity and power of the compressor by about one-third, but that this increases the velocity of discharge about 15 per cent. to the detriment of the mechanical efficiency of the machine.

The diagrams on page 212 will be of interest, as showing the distribution of heat and work.

Experimental Researches Necessary to Determine Precise Data for Gas Turbine Calculations.

We are now led to consider what experimental data are to be determined in order that we may substitute more or less precise computations for the hypothetical ones which have thus far been employed in our investigations.

Above all things it is desirable that a study should be made of the *work absorbed in the compression* and of the *final temperature of the compressed gases.* There appears to be no experimental difficulty in making these determinations. It is especially desirable that these studies should be made with rotary compressors, since the subject of piston compressors has already been well investigated.

Further, it is desirable that the phenomena of combustion should be studied, especially to determine:

1. The limits to the amount of combustible and air which can be used when attaining a perfect combustion at different pressures.

2. The *real* temperatures of combustion thus attained.

3. The conditions of temperature and pressure which will insure the self-ignition of the mixture.

4. The time required to enable a perfect combustion to be realized, and thus the volume of the combustion chamber to be determined.

5. The influence of these various elements upon the velocity of the gaseous current.

6. The influence of a greater or less quantity of vapor of water in the mixture. This should include the extent to which any portion of the water is dissociated, and under what conditions.

It is not difficult to see what methods might be employed to carry out such a programme. The present electrical appliances for determining high temperatures are already available for such a purpose.

It would then be desirable to investigate the subject of expansion. It is important to determine the extent to which the temperature is lowered by expansion. This is the most delicate part of all the investigations.

While it is comparatively easy to measure the temperature before expansion and the pressures before and beyond the nozzle, it is much more difficult to determine the true temperature of a gas leaving a nozzle at a high velocity.

The thermometric element placed in the midst of the gaseous current which is flowing at a velocity of 1200 to 1800 metres per second, takes a temperature higher than that of the gas, for the heat disengaged by friction is not dissipated by radiation. In every case it is necessary that the thermometric element should have a higher temperature than that of the surrounding medium in order that the heat thus disengaged may be removed and a stable condition secured.

Since we are ignorant of the value of the velocity (this involving a knowledge of the temperature of the gas), it is hardly practicable to be able to make a correction. It is, therefore, necessary to compute the temperature or the velocity by a series of successive approximations.

It seems preferable to leave aside any attempts to measure the final temperature of expansion, and to determine rather the kinetic energy of the gases leaving the nozzle. This may be done by measuring directly the push of the gaseous current upon a turbine blade, as has been done by Delaporte, Rateau, Stodola, and others.

Having thus determined the velocity of discharge w_2, we may deduce the temperature of the discharging gases from the formula:

$$\frac{w_2^2}{2g}=ECp(T_2-T_3).$$

We can then deduce a value of γ which takes into account the friction in the jet. Experiments of this kind will enable the best form of nozzle to be determined, as has already been done with the steam turbine, and especially to permit the question whether the form of constant acceleration proposed by Proell is preferable to the ordinary conical form.

Finally it is desirable to make an experimental determination of the value of the coefficient of transmission of heat from one gas to another through the walls of an assemblage of tubes, in order to aid in determining the dimensions of heat regenerators.

The Future of the Gas Turbine.

Having now discussed the question of the construction of the gas turbine we may take up the subject of the future in store for machines of this kind.

Their great theoretical interest has been apparent to all those who have examined these questions since the period when the success of the turbine of Laval demonstrated the value of the pressure type of steam turbine. This celebrated inventor follows the thought of Burdin and Tournaire, and suggested very early the idea of constructing a gas turbine. Many years have now passed, however, without the practical realization of this idea.

Other investigators have taken up the same idea, but thus far their efforts have not reached commercial success, while during the same period the steam turbine has emerged from the experimental workshop and acquired its well-known position among heat engines.

This should offer no reason for surprise, when we consider the multiplicity of technical difficulties which present themselves in the realization of a practical gas turbine. The success of the steam turbine, however, has elicited investigations of the greatest interest which lead us to approach the construction of a gas turbine without hesitation. Some investigations are yet required to enable the determination of the conditions of combustion and the exact laws governing the expansion. When these have been completed we will be in possession of all the data necessary for the turbine itself without guesswork.

Rotary compressors, multicellular blowers, turbine compressors of the Parsons, Curtis, and other types, are relatively further from a definite, practical solution, but everything leads us to believe that no material delay will occur in this direction.

We may thus expect to see commercially produced, a gas turbine, uniting in a certain degree the advantages of the gas engine and the steam turbine.

Without overlooking the inconvenience resulting from the presence of a compressor distinct from the motor itself, the gravity of this objection may be exaggerated.

If we are willing to accept the piston compressor (or use the alternative of the reduction of exhaust pressure below atmosphere) the gas turbine presents the same advantages of moderate bulk and weight which have made the success of the steam turbine.

The thermal efficiency of the new machine will be superior to that of the gas engine, but the lower mechanical efficiency of the gas turbine will reduce the total useful effect to about the same order as that of the Diesel motor; while motors using blast furnace gases should give an effective horse-power with an expenditure of 2000 calories.

It does not appear that any sensational invention can modify these results materially in the future. It is only by continual improvements in structural details that the mechanical efficiency may be increased by the reduction of mechanical losses.

The gas turbine will not be a universal panacea, neither will it dethrone the steam turbine. When we have to deal with the combustion of ordinary coal, nothing can surpass the steam boiler.

But for other combustibles, petrol, various hydrocarbons, alcohol, producer gas, furnace gases, etc., direct combustion is advantageous. It permits the avoidance of many important losses, and removes many operative objections and dangers.

The utilization of blast-furnace gases, coke-oven gases, etc., presents in itself an important field for the gas turbine, which may well replace the bulky engines now in use.

The gas turbine also appears to be as well adapted to the driving of dynamos and alternators as is the steam turbine. The same is true as regards the propulsion of ships.

It is also possible that the development of the gas turbine will permit the realization of motors of excessively light weight for use in aerial navigation.

We may thus predict for the gas turbine an extensive field of application, and it is altogether possible that practical experience will enable many special advantages to be developed, as so often has been the case in connection with the appearance of new and improved appliances.

CHAPTER IV.

THE DISCUSSION BEFORE THE FRENCH SOCIETY OF CIVIL ENGINEERS.—(*Continued.*)

THE paper of M. Sekutowicz, which has been given in full in the preceding chapter, naturally elicited an animated discussion which will be found in the memoirs of the Société.*

M. Réné Armengaud gave an account of his own experimental researches made at St. Denis in connection with M. Lemale, and these will be discussed at length in a following chapter.

M. Jean Rey discussed especially the problem of the compressor, showing the importance of the development of a satisfactory rotary or turbine compressor. To use a reciprocating compressor would be to deprive the gas turbine of most of the advantages to be gained over the ordinary gas engine.

Passing to the turbine compressor, M. Rey described the multiple turbine compressor of Rateau, as installed in the mines at Béthune, and constructed by Sautter, Harlé & Co.

In this machine there are four sets of turbine wheels arranged in series, revolving at 4500 revolutions per minute. The first set draws in the air at atmospheric pressure, and raises it to 1.7 kg. per square centimetre absolute (24 pounds per square inch). The second set increases the pressure to 2.9 kg. (41 pounds); the third to 4.9 kg., and the fourth to a final pressure of 7.2 kilogrammes absolute per square centimetre (102.4 pounds per square inch).

This compressor has a capacity of 1 kilogramme of free air per second; it has attained a capacity of 1.25 kilogramme, and the pressure has been pushed up to 8.2 kilogrammes absolute, or 7.2 kilogrammes above atmospheric pressure, or

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* Mémoires et Compte Rendu des Travaux de la Société des Ingenieurs Civils de France: May, 1906. Mm. Armengaud, Rey, Hart, Letombe, Bochet, Deschamps.

about 100 pounds per square inch over and above atmospheric pressure.

The efficiencies of the various sections differ, attaining 70 per cent. for the first, and 55 per cent. for the fourth; the mean efficiency of the entire machine being about 63 per cent.

M. Rey does not consider it practicable to construct such compressors to produce pressures of 30, 40, or 50 kilogrammes per square centimetre, as required by M. Sekutowicz, so that it would be necessary to supplement it by a small piston compressor.

M. Rey computes the practical efficiency of a turbine by calculating the energy absorbed in the compression of 1 kilogramme of air, as well as the energy developed by a kilogramme of burned gases upon the wheel, and his computation shows these two amounts to be about equal, so that there would be no power available for external use. This, however, hardly seems correct, since we have the energy furnished by the burned fuel added to that contained in the compressed air, and their sum should be considered. The practical operation of the turbine of Armengaud and Lemale also furnishes a refutation of the theoretical calculations of M. Rey, since it has developed 500 horse-power, only about one-half of which was required to operate the Rateau compressor by which it was served.

M. G. Hart called attention to the practical structural difficulties attending the realization of an operative gas turbine. In addition to the question of an efficient rotary compressor for high pressures, there are several other questions to be settled. Among these he emphasized the high rotative speeds to be realized, these bringing centrifugal stresses upon the materials of which the resistance would necessarily be reduced by the high temperatures. Even if the difficulties attending the cooling of the rotating parts are successfully overcome, there will be expansion and contraction stresses which must be taken into account.

As regards the combustion chamber there are several questions involved in its successful construction, although M. Armengaud appeared to have adopted an effective design. M. Hart suggested that several combustion chambers arranged in series might be found more advantageous than a single one of larger size, especially in connection with speed regulation for light loads.

The practical solution of the gas turbine question, according to M. Hart, appears to lie in the perfection of a number of details, a result attainable only by means of exhaustive experimental investigations.

M. Bochet called attention to the fact that high degrees of compression were necessary if high thermal efficiencies were to be attained, citing the experience of the Diesel motor, in which the temperature of compression is sufficient to cause the ignition of the combustible. Such high compressions, however, are as yet entirely beyond the powers of the best turbine compressors, a fact which militates severely against the success of the gas turbine so far as efficiency is concerned.

M. L. Letombe compared the possibilities of the gas turbine with the achieved performances of the piston gas engine. He believed that the steam turbine had, in some cases, been found preferable to the steam engine because of its greater simplicity, but it seemed as if this point could not be advanced for the gas turbine, because the latter machine, at least so far as developed at present, was more complicated than the reciprocating gas engine.

In closing the discussion, M. Sekutowicz reviewed the criticisms which his paper had elicited, commenting upon the influence which the variability in the specific heat of gases at very high temperatures might have upon his computations, and emphasizing the desirability of submitting the doubtful points to the test of actual investigation in the mechanical laboratory.

CHAPTER V.

ACTUAL BEHAVIOR OF GASES IN NOZZLES.

ONE of the most essential elements in the success of the gas turbine lies in the practicability of the conversion of the original potential energy of the gases into kinetic energy in the nozzle. The extent to which this can be accomplished is yet a matter for discussion.

Experimental investigations upon the free expansion of gases in nozzles, conducted by Dr. Charles E. Lucke, at Columbia University, appear to show that the nozzle is a far less efficient means for the conversion of energy than the piston and cylinder. Referring to experiments made upon the expansion of compressed air to show the extent of temperature drop, Dr. Lucke says:

"Holding a thermometer in the stream of air issuing from an open valve or nozzle on a compressed air main will show, for even a pressure drop of 100 pounds per square inch, only three or four degrees temperature change. This also may be due to impact on the thermometer raising the temperature of the moving gases by bringing them to rest on the bulb; but again this will not account for the whole difference between what is observed and what would be were this free expansion equivalent to balanced expansion. To eliminate the errors of impact as much as possible, a thermal couple stretched axially along the jet and made of fine wire has been used by the author for a measurement of the temperature of the air when moving at the maximum velocity. The maximum temperature drop for air under 100 pounds initial pressure, expanding through a steam turbine nozzle into atmosphere, is only 30 degrees F. This result is only 12 per cent. of the temperature drop that would have resulted did the air suffer balanced expansion without gain or loss of heat.

"Another instance of the same lack of equivalence in results by free and balanced expansion is found in the experiments of Tripler and Linde on the making of liquid air. In this work air highly compressed (2000 to 3000 pounds per square inch) is first cooled by water and then some of the air freely expanded through a hole, the discharge passing around the pipe feeding the hole. This was intended to cool the air in the pipe lower than the critical temperature for liquefaction under the high pressure used. The results were enormously different from the case for balanced expansion, the temperature drop through the nozzle being about ¼ degree F. per atmosphere-pressure drop, according to one report. More accurately the results for the Linde process are shown in the following table, the initial pressure being 220 atmospheres.

Temperature approaching the nozzle.	Actual temperature drop through nozzle.
+ 30° F.	35° F.
0° F.	65° F.
— 30° F.	80° F.
— 60° F.	96° F.
—100° F.	112° F.
—150° F.	135° F.

"Unless, by an increase of knowledge of free expansion of perfect gases, it becomes possible to produce results equivalent to those obtained with balanced expansion, there cannot be the same amount of heat transformed into work by the gas turbine engine as by the cylinder-and-piston gas engine."

Later investigations made by Dr. Lucke in operating a De Laval steam turbine with compressed air gave interesting results, an abstract of which is here given.

"For convenience of operation the air was cold air, whereas in the practical gas turbine the air would be hot and possibly more or less mixed with steam, or possibly no air at all, but carbon dioxide. In any event, the working fluid would be largely a perfect gas. The turbine used was a De Laval standard 30 horse-power machine intended for steam at 110 pounds pressure and having six nozzles. The turbine wheel runs at 20,000 revolutions per minute, and the power shaft 2000 revolutions. The air used for driving the turbine was measured by a Westinghouse metre. The tests were run on no load, because the compressor used was not sufficiently large to supply the amount of air needed at full load, or even at full speed without load. With each type of nozzle three different initial pressures were used, each with a different number of nozzles. Readings were taken of the temperature of the air entering the turbine and the temperature of the air in the exhaust chamber, with the corresponding pressures. The nozzles fitted to this turbine in holes Numbers 1 and 4 are 110 pounds pressure and 25½ inches vacuum; in holes Numbers 3 and 6 for 110 pounds pressure and 26.3 inches vacuum; and in holes Numbers 2 and 5 for 110 pounds steam pressure and atmospheric exhaust. The results of the pressure-drop runs are given in the following table, which also gives the theoretical temperature-drop, assuming an adiabatic expansion of air between the same pressures.

"From this it appears that the temperature-drop realized varies from 4 to 18 per cent. of the theoretical or adiabatic temperature-drop. The preceding results are given with respect to speeds also, which varied from 520 to 1920 revolutions per minute. To show according to what law this complete process takes place, the exponent of the temperature ratio in the equation between pressure ratio and temperature ratio, which for adiabatic expansion of air is .29, was determined and found to lie between .1005 and .0380."

These results obtained by Dr. Lucke must be compared with the practical ones secured by the engineers of the *Société des Turbomoteurs* at St. Denis and the experiments made by M. Alfred Barbezat upon the small experimental

Results of Temperature-drop Runs with De Laval Turbine, working with Compressed Air.

Pressure in pounds gauge. Temperatures in degrees Fahrenheit.

Initial		Final Pressure.	Observed temperature.	Final pressure.	Theoretical temperature.	Temperature observed.	Drop, theoretical.	Per cent. of theoretical, realized.	Speed	Exponent	
Pressure.	Temp.									Theor.	Observed.
85	98	.03	58	.03	—123	40	221	18.1	1920	.290	.0380
55	93	.02	89	.02	—103	24	201	11.9	1560	.290	.0369
24	96	.01	83	.01	— 40	13	136	9.5	950	.290	.0240
54	97	.10	81	.10	—102	18	201	8.5	1880	.290	.0198
40	102	.08	86	.08	— 76	17	178	9.5	1630	.290	.0232
24	109	.03	93	.03	— 30	16	139	11.5	1200	.290	.0262
35	108	.08	91	.08	— 61	17	169	10.	1695	.290	.0252
23	104	.05	95	.05	— 30	9	134	6.7	1290	.290	.0156
12	101	.03	96	.03	— 1	5	100	5.	855	.290	.0169
123	85	.01	60	.01	—176	25	261	9.6	1540	.290	.0206
92	91	.01	62	.01	—150	29	241	12.	1270	.290	.0258
60	91	.00	69	.00	—116	22	207	11.6	1940	.290	.0251
95	83	.01	51	.01	—157	26	240	10.8	1750	.290	.0240
76	93	.02	66	.02	—134	27	227	11.8	1565	.290	.0282
58	92	.02	68	.02	—113	24	205	11.7	1285	.290	.0289
75	81	.02	59	.02	—140	22	221	10.	1710	.290	.0226
54	89	.01	70	.01	—109	19	198	9.6	1410	.290	.0232
35	92	.01	74	.01	— 72	18	164	11.	1075	.290	.0268
68	88	.02	48	.02	—128	40	216	18.5	1855	.290	.0420
52	89	.01	59	.01	—106	30	195	15.4	1660	.290	.0374
32	87	.01	68	.01	— 69	19	156	12.2	1220	.290	.0380
42	86	.09	63	.09	— 90	23	176	13.	1670	.290	.0382
31	89	.07	78	.07	— 65	11	154	7.2	1505	.290	.0152
17	87	.03	82	.03	— 23	5	110	4.6	1040	.290	.0130
-123	-88-	-.14	-68-	-.14	—173	20	261	7.7	1140	.290	.0160
91	93	.09	76	.09	—148	17	241	7.1	880	.290	.0155
61	91	.05	80	.05	—120	11	211	5.2	530	.290	.0122
103	91	.30	76	.30	—157	15	248	6.1	1300	.290	.0133
68	93	.22	81	.22	—124	12	217	5.5	910	.290	.0117
48	90	.12	82	.22	— 98	8	188	4.3	520	.290	.0105

turbine of Mm. Armengaud and Lemale show a much greater drop. It is greatly to be desired that this whole subject of free expansion in nozzles for steam, for air, and for mixed gases, at various temperatures should be thoroughly investigated experimentally, and it might well occupy the efforts of some of the highly equipped mechanical and physical laboratories of the technical schools.

CHAPTER VI.

THE PRACTICAL WORK OF ARMENGAUD AND LEMALE.

THE most complete account of the Armengaud and Lemale turbine, the gas turbine which, by its practical performances has done the most to demonstrate that the gas turbine is a reality, and not merely an academic discussion, is contained in an article by the late M. Réné Armengaud, published in Cassier's Magazine, and here reprinted.* M. Armengaud reviews the principles of the gas turbine, and describes some early devices, and then proceeds:

Heat motors in general service at the present time may be grouped into the following classes:

1. Alternating steam motors (reciprocating steam engines).
2. Alternating combustion motors (reciprocating gas engines).
3. Continuous steam motors (steam turbines).
4. Continuous combustion motors (gas turbines).

Of these various machines the latest, and certainly the least known, is that which appears to have a most interesting future, the gas turbine, and it is this which I now propose to discuss.

A successful gas turbine aims to combine the great advantages of the gas engine, including the elimination of the steam boiler and a high thermal efficiency, with the special advantages of the steam turbine, *i. e.*, simplicity of construction, lightness, and the greatly desired property of continuous motion in one direction, with the accompanying features of control and regulation.

The various plans which have been discussed for the de-

* The Gas Turbine. Practical results with actual operative machines in France. By Réné Armengaud, Cassier's Magazine, January, 1907.

sign of the gas turbine may be divided into three groups: hot-air turbines, explosion turbines, and combustion turbines.

So far as the first group is concerned, I have not attempted to make any investigations in this direction, believing this system to offer few advantages. The only machine of this kind of which I have any knowledge is that of Dr. Stolze, of Charlottenburg, of which the following description is abstracted from his patents. Air is compressed by means of a helicoidial compressor to about 1½ atmospheres. The air, after having circulated about a furnace, expands, and is then passed through a turbine attached to the same shaft as the compressor.

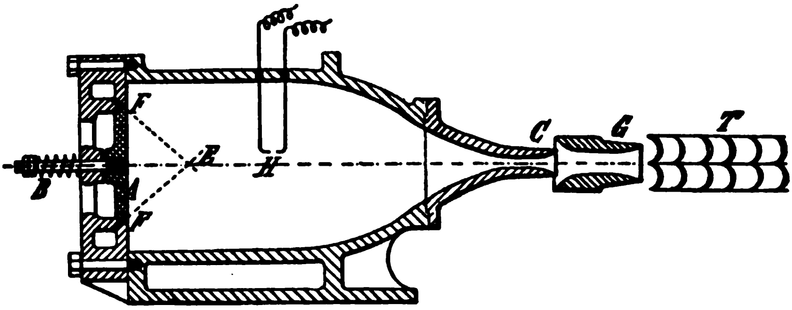

Fig. 74.—General arrangement of explosion gas turbine.

In the case of the second group, the explosion turbines, the air compressor is eliminated or greatly simplified. The explosive mixture is formed in the same chamber in which it is ignited, being either at atmospheric pressure or slightly above, and by its expansion, consequent upon the explosion, it acts upon the turbine wheel. The principle of such a machine is shown in Fig. 74. The explosion chamber is closed at the back by a valve *A* held to its seat by a light spring *B*, the chamber having an expanding nozzle opening at *C*. The gas enters at small openings, as at *E* under the seat of the valve, and the air is admitted at *F*, the mixture being ignited electrically at *H*, and discharged through the nozzle C upon the buckets of the turbine wheel T. The discharged gases pass through an induced current nozzle G which acts to reduce the temperature of the issuing gases and lower the velocity of the jet as it acts upon the turbine wheel.

Such an apparatus, when properly proportioned, will make about three explosions per second, and will continue to run automatically after it has once been started.

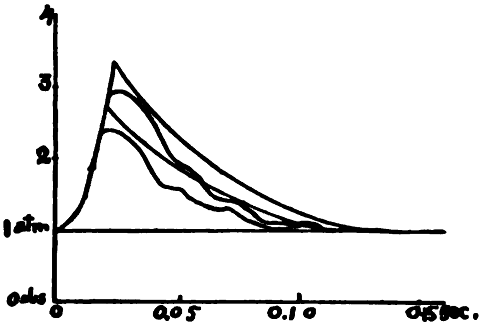

FIG. 75.—Explosion turbine diagram.

Various theories have been advanced to explain the action of this device. The most satisfactory explanation of the periodic action is that of the sudden cooling of the chamber after each explosion. This cooling causes a corresponding drop in the pressure, followed by the opening of the valve A and the aspiration of the air and gas, and as soon as the explosive mixture reaches the igniter a fresh explosion follows. In Fig. 75, the variations in pressure in the chamber are shown as a function of time. The maximum effective pressure ranges from 2 to 3 kilogrammes per square centimetre, or about 30 to 45 pounds per square inch, although the theoretical pressure in such an open vessel should reach 4 kilogrammes, so that the mixture of the gas and air is probably imperfect.

Theoretically, the explosion turbine should have a certain thermal advantage over the corresponding cycle for a combustion turbine. The specific heat at constant volume being lower than the specific heat at constant pressure, the same quantity of heat acting upon the same mass of gas should produce a higher temperature after the explosion than after a combustion. Since, according to the principle of Carnot, the efficiency is proportional to the maximum temperature of the fluid before expansion, the explosion turbine should be more efficient than the combustion turbine.

Unfortunately the high velocities of discharge of the gases, and the variations in the pressure, render it impracticable to realize more than a small fraction of the energy of the jet upon the wheel. Thus, the theoretical efficiency of such a

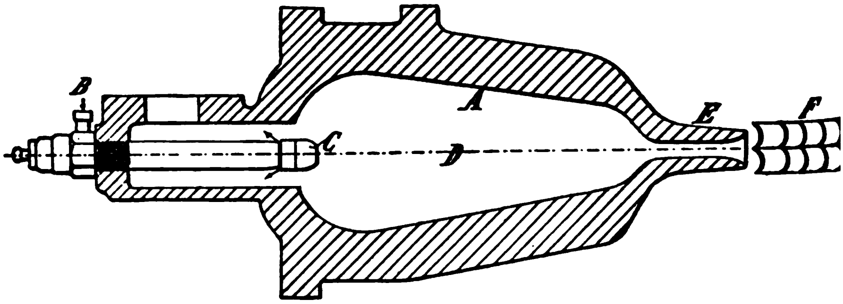

FIG. 76.—Combustion gas turbine. *A*, combustion chamber. *B*, fuel inlet. *C*, fuel sprayer. *E*, expansion nozzle. *F*, turbine.

machine should be about 16 per cent., while the actual performance does not exceed 3 to 4 per cent. In addition to this defect there are operative difficulties with the springs and valves, and the frequent breakages and delicate adjustments have rendered experiments to improve the apparatus unsatisfactory.

There remains, then, the combustion turbine, which, in spite of the necessity for an air compressor, is greatly to be preferred, especially for large units. This machine consists in principle of a combustion chamber *A* Fig. 76. supplied by a continuous current of compressed air, and also by a continuous supply of liquid fuel, gasoline, petroleum, or the like, under pressure through a tube B, the mixture being ignited at the start by a platinum wire C, the combustion developing a constant temperature of about 1300 degrees C. in the chamber D. The fluid products of combustion are then continuously discharged through a nozzle E, upon the buckets of the turbine wheel F.

The principal defect in this apparatus in comparison with the reciprocating gas engine is the necessity for a separate air compressor, instead of having the compression of the charge effected in the motor itself. This defect is partially remedied by the diminution of the losses through the walls, and by the possibility of an expansion which is practically

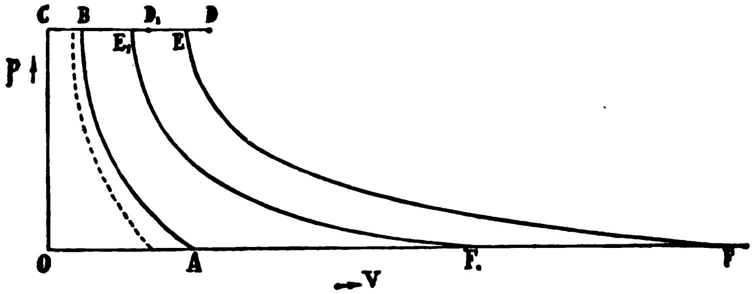

Fig. 77.—Diagram for combustion turbine.

adiabatic. The combustion is also more complete than is possible in a working cylinder, and all the products of combustion are utilized.

The action of a combustion turbine is graphically shown in Fig. 77. In this diagram the area $OABC$ represents the energy required for the air compressor. The combustion of the liquid fuel increases the volume from CB to CD. If any vapor of water is introduced, this volume will be diminished from CD to CD_1, while at the same time its mass increases the volume of CD_1 to CE. The effective energy exerted by the turbine will be represented by the area $OFEC$ and that available after the deduction of the work of compression will be $AFEB$.

In endeavoring to produce such a cycle in an actual working machine, the following practical difficulties must be overcome:

A gaseous fluid moving at a high velocity must be kept constantly ignited, by a device which must not be affected by the high temperature of the combustion chamber.

The mixture of the combustible and the air must be made as perfect as possible.

The injurious action of the gaseous products at a high temperature upon the parts of the apparatus, and upon the turbine wheel itself, must be prevented.

For three years a machine complying with these conditions has been running successfully in the shops of the Société des Turbomoteurs at Paris, this apparatus being the Armengaud-Lemale turbine, of which some further description will be given.

The original machine was made from a De Laval steam turbine of 25 horse-power, arranged to be operated with compressed air instead of steam. The air was supplied at any desired pressure from a high speed compressor, of which the efficiency had been closely determined, while prolongations of the pipe which connected the compressor to the turbine formed the combustion chamber. At the entrance of each chamber the gasoline, mixed with the air, was ignited by an incandescent platinum wire, this ignition being necessary only at the starting of the operation, the combustion being maintained continuously thereafter at constant pressure. The combustion chambers were lined with refractory material, and a temperature of about 1800 degrees C. was produced. In order to reduce the temperature to practical limits the chamber was cooled by the introduction of vapor of water generated in a spiral imbedded in a portion of the combustion chamber. The steam thus produced was allowed to mingle with the gases of combustion before expansion in such proportion that the temperature of the mixture was about 400 degrees C.

Although this apparatus was necessarily crude and not proportioned in such a manner as to give the best results, it enabled the conditions essential for a good efficiency to be determined.

Among the practical points thus determined were proofs that it was entirely possible to maintain the combustion chamber, turbine wheel, and fuel pulverizer in operative condition. The experiments also showed it to be practicable to maintain a very high temperature continuously in the actual combustion chamber, and, by means of this high heat to secure a perfect combustion of any combustible. The work of compression having been carefully ascertained for the purpose of deducting it from the brake power developed by the entire machine, it appeared that even with this imperfect apparatus the total power was about double that necessary to drive the compressor. This result was attained with a pressure of about 10 kilogrammes per square centimetre, and a temperature of 400 degrees C. at the exhaust.

As has already been said, the excessively high temperatures developed were reduced in the earlier experiments by mixing a certain quantity of steam with the gases of combustion before expansion. This method, while accomplishing the result desired, also acted to lower the efficiency of the turbine, doubtless because of the latent heat of vaporization lost in the exhaust. In the diagram, (Fig. 78) the curves show the manner in which the economical performance of this machine varied, represented as a function of the upper pressure and of the temperature of the exhaust gases. This diagram has been computed upon a basis of 60 per cent. efficiency of the turbine wheel, and 80 per cent. of the compressor. For example, with a pressure of 30 kilogrammes per square centimetre, and an exhaust temperature of 450 degrees C., an efficiency of 18 per cent. is obtained.

It thus appears that the efficiency depends both upon the pressure and upon the temperature of the exhaust gases.

In order, therefore, to obtain the best efficiency it is necessary to prevent cooling the gases before expansion, either by introducing steam into the combustion chamber, or otherwise, and to effect the greatest possible reduction in temperature in the expansion alone.

The difficulties accompanying the high temperatures may be met in the case of the combustion chamber and other fixed parts by the use of a water jacket and by the employment of a refractory lining, and the real difficulties are reached only when it becomes necessary to provide for the effect of the highly heated fluid upon the rotating metallic wheel, already weakened by the heavy centrifugal stresses to which it is necessarily subjected.

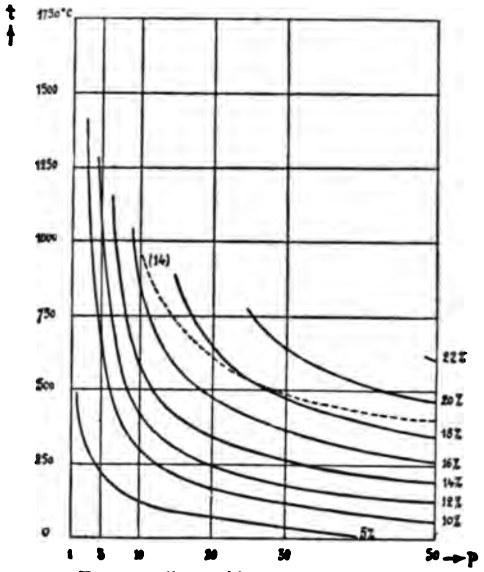

FIG. 78.—Gas turbine economy curves.

The most practical way of keeping the turbine wheel cool is to follow the jet of hot gases by another jet of a low temperature so that the buckets of the wheel pass successively through alternate hot and cool zones, the average temperature of the two jets being sufficiently low to prevent injury to the metal. The low temperature jet found most practicable is that of low pressure steam, and this is readily provided from the water jacket and from a device arranged as a regenerator in connection with the exhaust gases.

This arrangement, shown in Fig. 79, gives a general idea of the system. The air from the compressor enters at D and is mixed with the liquid fuel in the concentric nozzle EE and

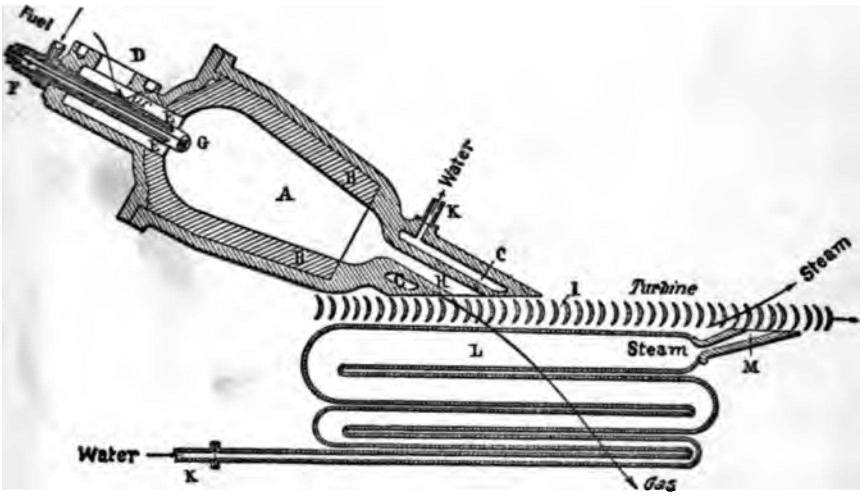

FIG. 79.—Mixed gas and steam turbine. Air enters at D, fuel at F, the ignition is made at G. The combustion chamber A is lined with carborundum. The nozzle H is water-jacketed, and the hot water passes to the steam generator L, which is heated by the exhaust gases from the turbine. The steam acts to propel and cool the wheel by the nozzle M.

ignited by the platinum wire at G. The combustion takes place continuously at constant pressure in the chamber A, and the products of combustion are discharged through the expansion nozzle H upon the buckets of the turbine wheel I. The nozzle itself is protected by a water jacket C, the water leaving the jacket at K. On the other side of the wheel there is arranged a sort of flash steam generator L, this being composed of a serpentine pipe of continually increasing diameter, the water entering the small end at K, this entrance forming a part of the discharge pipe from the water jacket of the nozzle *H*. The steam generator *L* is placed in the path of the exhaust gases leaving the turbine wheel, and these highly heated gases furnish the heat necessary to convert the water into steam, the vapor thus produced being discharged through the nozzle *M* upon the turbine wheel, thus acting both to aid in the propulsion and to form a zone of comparatively low temperature to abstract heat from the wheel. By this arrangement it is possible to reduce the temperature of the wheel to practicable limits, provided the temperature of the exhaust gases is sufficiently high to produce enough steam. That is, the expansion of

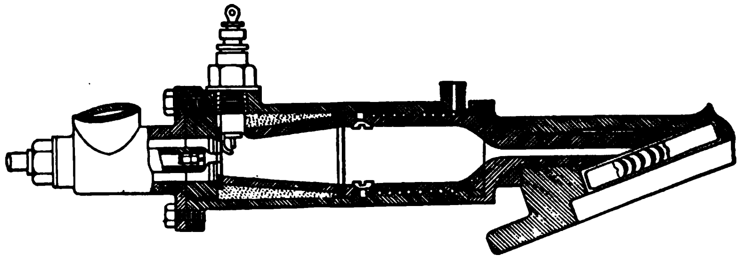

FIG. 80.—The Lemale combustion chamber.

the gases in the nozzle must not lower their pressure and temperature so far as to keep down the volume of steam too low. In such case it is always practicable to admit a small quantity of superheated steam into the combustion chamber and thus obtain the required temperature without affecting the efficiency of the machine too much.

The general heat balance of a gas turbine using a steam regenerator according to the above plan is shown in Fig. 81, in which the total quantity of energy produced by the fuel is represented by the dimension X, and the various losses indicated by the cross hatched portions in the body of the diagram. The efficiency of the machine is then obtained as the ratio $Y : X$, Y being composed of two parts, one of which is obtained from the action of the gases upon the wheel and the other by the steam.

In this arrangement the expansion of the gases and the steam occur in parallel, so to speak, this being clearly indicated in the diagram. For constructive reasons, however, it is found convenient to adopt the previous system, the steam produced in the regenerator being delivered into

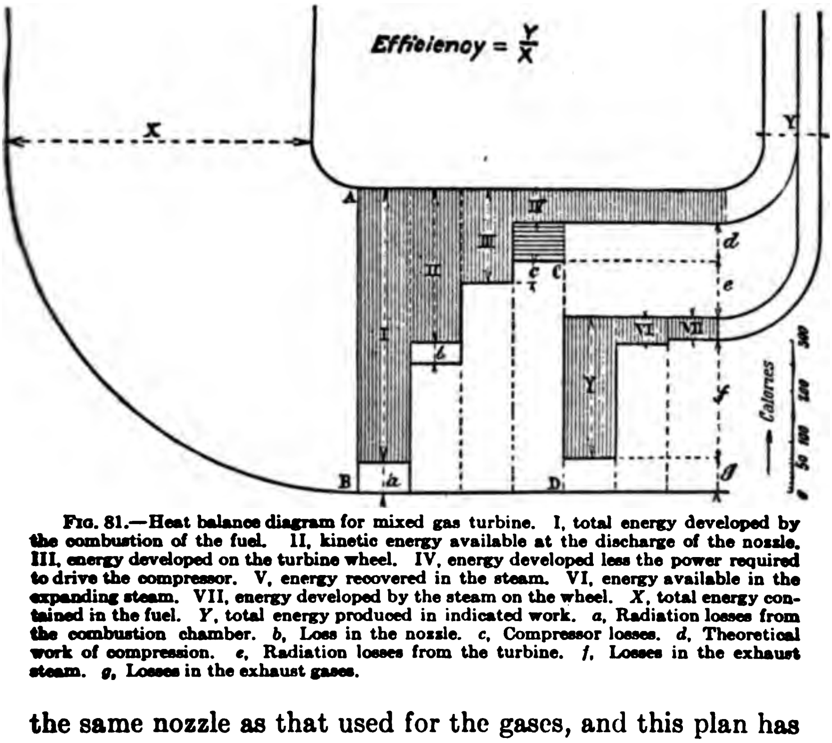

FIG. 81.—Heat balance diagram for mixed gas turbine. I, total energy developed by the combustion of the fuel. II, kinetic energy available at the discharge of the nozzle. III, energy developed on the turbine wheel. IV, energy developed less the power required to drive the compressor. V, energy recovered in the steam. VI, energy available in the expanding steam. VII, energy developed by the steam on the wheel. *X*, total energy contained in the fuel. *Y*, total energy produced in indicated work. *a*, Radiation losses from the combustion chamber. *b*, Loss in the nozzle. *c*, Compressor losses. *d*, Theoretical work of compression. *e*, Radiation losses from the turbine. *f*, Losses in the exhaust steam. *g*, Losses in the exhaust gases.

the same nozzle as that used for the gases, and this plan has been adopted in our most recent turbine, even at some reduction in the thermal efficiency.

This machine, shown in several views, is of the same general type as the Curtis steam turbine, and is capable of delivering from 400 to 800 h. p., according to the compressor capacity utilized. The turbine is operated at 4000 revolutions per minute, and the speed regulation is effected by a throttling valve in the air admission pipe for small speed variations, and by a change in the fuel supply for larger changes, the regulating valves being controlled by a Hartung governor. There are three pumps attached to the machine, one the air compressor, another for the fuel supply, and the third for the water.

The combustion chamber is made of cast iron, lined with carborundum, the cast iron being protected with a water jacket. An elastic non-conducting lining is placed between the carborundum and the outer shell, this providing a bedding for the carborundum and also permitting a slight movement for differences in expansion and contraction. The extremity of the chamber and the nozzles are surrounded by a jacket space in which the water and steam circulate, the nozzles being of the expanding type similar to those of the De Laval steam turbine, although the expansion is completed in a shorter time. It is necessary that the expansion should be effected in a single operation in order that the temperature be sufficiently reduced before the gases reach the wheel.

The gasoline or other liquid fuel is delivered into the combustion chamber through a pulverizer, or atomizer, the construction of which is shown in Fig. 86. This is arranged with a reverse annular opening *B* delivering the fuel backward against the incoming stream of air, the angle causing the gasoline to form a sort of cone of minute particles, these becoming ignited as soon as their decreasing velocity permits. The preheating of the fuel also renders the ignition easy. The atomizer is protected against the intense radiant heat of the chamber wall by the current of air with which it is continually surrounded. The igniting coil of platinum wire *D* is protected by a steel cap *C*, the electric current entering by the central insulated rod *E*, the circuit being completed through the machine itself. A pressure of 2

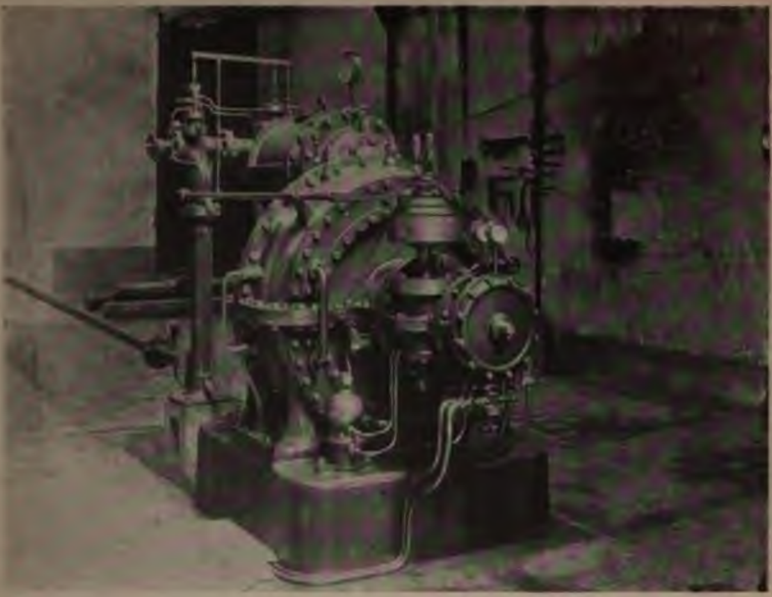

FIG. 82.—The Armengaud and Lemale gas turbine. A view of the 500 horse-power turbine in the experimental laboratory at St. Denis.

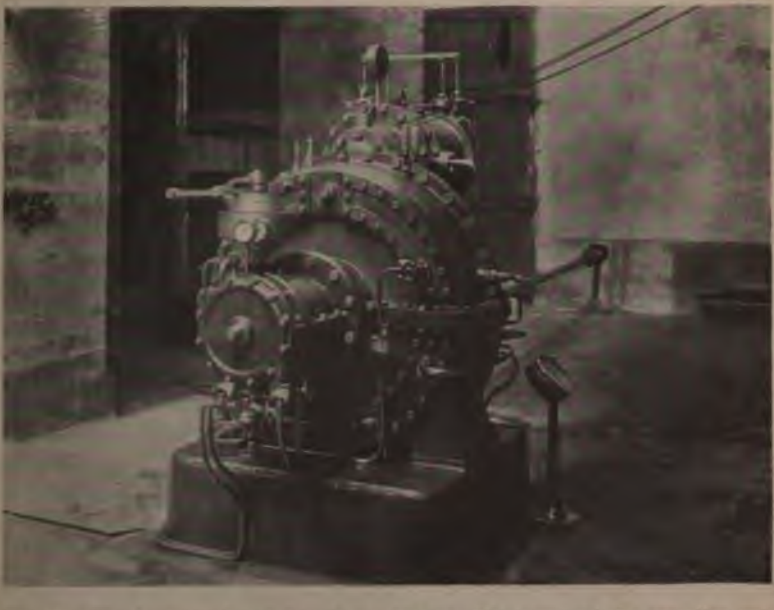

FIG. 83.—The Armengaud and Lemale gas turbine. This view and the preceding one show the wheel casing, governor, and general arrangement.

FIG. 84.—The Armengaud and Lemale gas turbine. This view shows the combustion chamber, air and fuel inlets and connections.

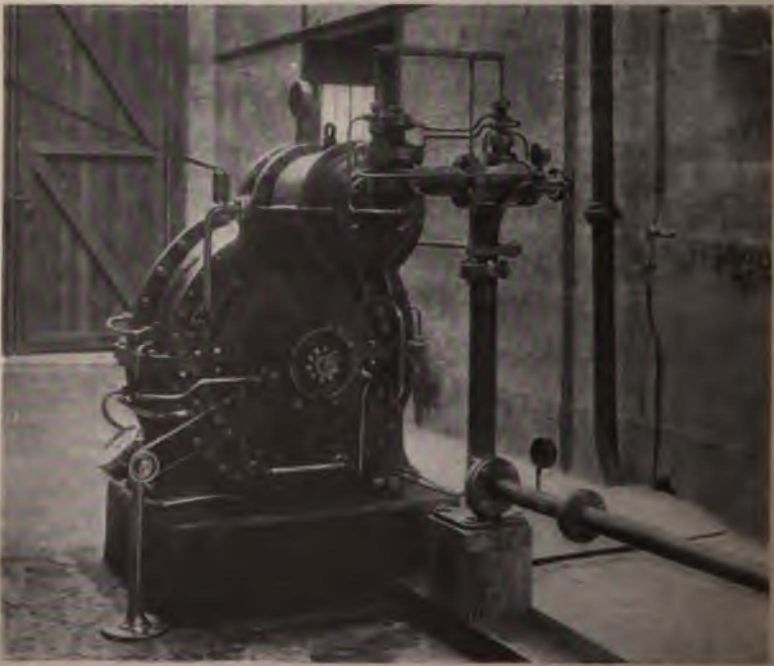

FIG. 85.—The Armengaud and Lemale gas turbine. Another view of the combustion chamber, with air and fuel connections.

volts is found sufficient to render the platinum wire incandescent. The atomizer is inserted into the combustion chamber in such a manner that it can readily be removed for inspection and cleaning, this operation also giving complete access to the chamber itself.

The turbine wheel is arranged to be cooled by water circulation as shown in Fig. 87, this representing a section of the rim and a portion of the disc. *A* and *B* are circular channels in the body of the rim, these being supplied with water by radial passages as at *E*. Small passages also permit the water to enter into each blade of the turbine and the

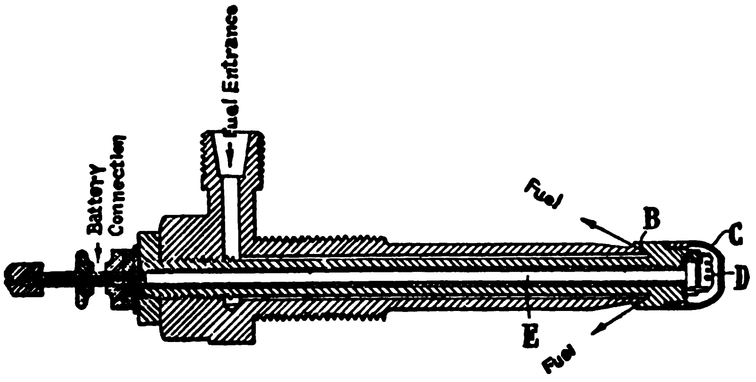

FIG. 86.—Section of pulverizer and igniter.

difference in specific gravity between the hot and cold water is found to make an automatic circulation, in connection with the centrifugal force due to the high velocity of rotation.

The air supply for the turbine is furnished by a polycellular rotary compressor of the Rateau system. This important adjunct to the gas turbine, shown in Fig. 88, is composed of a number of turbine blowers arranged in series and especially designed to be operated at very high rotative speeds, so that it may be directly connected to the gas turbine.

The importance of the compressor is second only to that of the turbine itself, since it is of little value to possess a rotary combustion motor if a reciprocating compressor is a necessary auxiliary. It is on this ground, more than almost any other, that the design of a successful gas turbine has been considered problematical, and here, as in many other cases in the

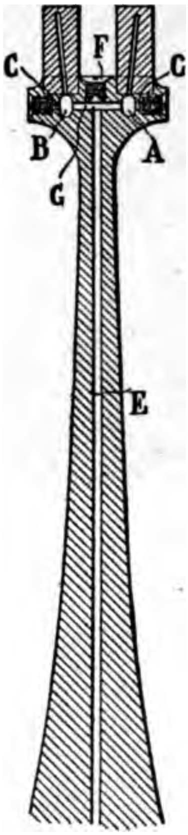

FIG. 87.—Section of wheel of gas turbine, showing passages for cooling-water.

history of the development of a device, the progress of other departments of work becomes essential to complete success. The work of M. Rateau in the improvement of the steam turbine is well known, and by the application of the experience thus gained, a machine for the supply of compressed

Fig. 88.—The Rateau polycellular air compressor. This multiple turbine blower furnishes air at more than 100 pounds pressure, with an efficiency of over 65 per cent., and is being used in connection with gas turbine researches.

air to the gas turbine has been produced, involving only rotary motion, and thus capable of being driven directly by the turbine, and having an efficiency sufficiently high to permit a good performance of the combined apparatus.

The Rateau compressor is practically a reversal of the steam turbine, and is composed of a number of elements connected in series, so that the pressure is cumulative, the action being similar to that of the multiple centrifugal pumps which have been employed with such success for delivering water against high heads.

Each element of this compressor consists of two parts, the revolving wheel and the diffuser. The diffuser is arranged to provide discharge passages for the air, having gradually increasing section for the flow, in order that the velocity of the air as it leaves the wheel may be reduced with the least possible loss, the kinetic energy being converted into pressure. The length of the machine is such that intermediate bearings have been introduced to provide support and stiffness to the rotating parts, and the whole design of the compressor has been so carefully worked out that an efficiency of 65 per cent. has been already attained, and present experiments indicate that this performance will be surpassed.

In Fig. 88 a Rateau compressor of three sections is shown, but larger machines have been constructed, and the pressures attained naturally depend upon the number of sections. Experiments have shown that in the first section the air is compressed to 1.7 kilogrammes per square centimetre, or about 24 pounds per square inch, absolute, the succeeding pressures being 2.9 kilogrammes, 4.9 kilogrammes, and 7.2 kilogrammes per square centimetre, the latter corresponding to 112 pounds per square inch above vacuum.

In a subsequent issue of Cassier's Magazine,* an article was published containing a communication from M. Alfred

* Cassier's Magazine, April, 1908.

Barbezat, who had been associated with M. Réné Armengaud, and who continued in the work after the death of the latter.

M. Barbezat reviewed the early work of M. Armengaud, and then described the later progress as follows:

The general principle of the machine involves the delivery of air under pressure into a pear-shaped chamber lined with refractory material and provided with an expanding nozzle through which a uniform flow of gases can be delivered upon the blades of the wheel. In the centre of the air nozzle there is arranged an axial tube, with a pulverizer at the inner end, through which the fuel, in the form of gasoline, or similar liquid hydrocarbon, is forced into the chamber. The electric sparking device enables the fuel to be ignited on starting, after which the high temperature of the chamber maintains the combustion indefinitely. The high temperature produced by the combustion greatly increases the volume of the air, and this, together with the gaseous products of the combustion of the fuel, flows at a high velocity through the expanding nozzle upon the blades of the wheel.

In dealing with such high temperatures, the temperature of the combustion being about 1800 degrees C., the best refractory lining for the combustion chamber has been found to be carborundum, this being a product of the electric furnace, and thus having already sustained even higher temperature than those in the turbine combustion chamber. An elastic backing of asbestos provides for the expansion of the carborundum lining, and the nozzle through which the gases are discharged upon the wheel is also made of carborundum.

In addition to the provision of a refractory lining, it has been found necessary to surround the combustion chamber with a water jacket in the form of a coil of pipe imbedded in the metal of the chamber walls, much in the same manner as such coils are used in the tuyeres of blast furnaces, and the circulation of the water in the coils aids in keeping the

9.—The 30 horse-power experimental gas turbine of the Société des Turbomoteurs.

FIG. 90.—The 300 horse-power gas turbine of Armengaud and Lemale connected to the Rateau polycellular turbine compressor.

temperature of the combustion chamber walls within practicable limits.

After the water has circulated in the jacket tube it is delivered, through small holes, into the gases just before they enter the nozzle, and is there converted into steam, this acting both to lower the temperature of the issuing gases to a point where they will not injure the blades of the turbine, and also itself being discharged upon the wheel with the gases and forming a part of the jet, which is thus composed of mingled gas, steam, and highly heated air.

In order to obtain the desired result of a machine involving only rotary motion, it is necessary that the compressed air by which the combustion chamber is fed should be produced, not by a reciprocating piston compressor, but by some form of rotary machine, preferably so arranged that it can be coupled directly to the turbine itself. This means that the complete gas turbine must also include a rotary air compressor, and that such a compressor must have a high efficiency in itself, otherwise it will produce such a large proportion of negative work as to detract materially from the efficiency of the combined machine, even though the actual thermal efficiency of the turbine be high.

After a number of experiments upon single impeller turbine air compressors, driven at high rotative speeds by De Laval steam turbines, the services of Professor Rateau were enlisted in the work, and a multiple turbine compressor, designed by him especially for this work, was constructed at the works of Brown, Boveri & Co., at Baden, Switzerland. This machine is arranged in three sections and provided with continuous cooling circulation, and, being thoroughly tested, was found to be capable of delivering one cubic metre of air per second at a pressure of 6 to 7 atmospheres, with an efficiency ranging between 60 and 70 per cent.

The illustration (Fig. 90) shows the arrangement with this compressor coupled directly to the large experimental turbine constructed by M. Armengaud, the turbine and the compressor thus forming practically one machine.

In this arrangement the compressor was found to absorb about one-half the total power developed by the turbine, the machine, when running at about 4000 revolutions per minute developing about 300 horse-power over and above the negative work absorbed by the compressor. At the present time experiments are being made upon the thermal efficiency of the machine, which is, as yet, not as high as that of the reciprocating gas engine; but these tests are not yet completed, and the results not available for publication.

During the past few months a practical application of this turbine has been made in connection with the operation of submarine torpedoes.

It is well known that in certain types of such machines the motive power for the brief period which elapses between the discharge and the contact with the target is derived from a store of compressed air, and in some such torpedoes the compressed air acts upon a turbine wheel similar to the steam turbine. This principle has now been extended to the use of the gas turbine, the compressed air from the reservoir passing through a combustion chamber and the total products of combustion together with the vapor of water acting on the turbine, and its capacity thus increased over that operated by compressed air alone.

The turbines made for this purpose develop 120 horse-power at a speed of 1000 revolutions per minute, the expansion ratio being 8.4. The weight of the turbine alone is 73.16 kilogrammes, or about 1.3 pounds per horse-power. Including the weight of the reservoir of compressed air, together with the petrol and water for a discharge lasting 80 seconds, the total weight of the whole apparatus is about 295 kilogrammes, or a little less than 2.5 kilogrammes, or 5.5 pounds per horse-power.

Although the gas turbine is, therefore, still in the experimental stage, it has made material advances in the past year, the 300 horse-power combined compressor and turbine being an accomplished fact, and a number of 120 horse-power machines of a special type being actually installed in submarine torpedoes completed for active service. When this rate of progress is compared with the time required to bring the reciprocating gas engine to its present state of perfection, there appears to be reason for encouragement and interest.

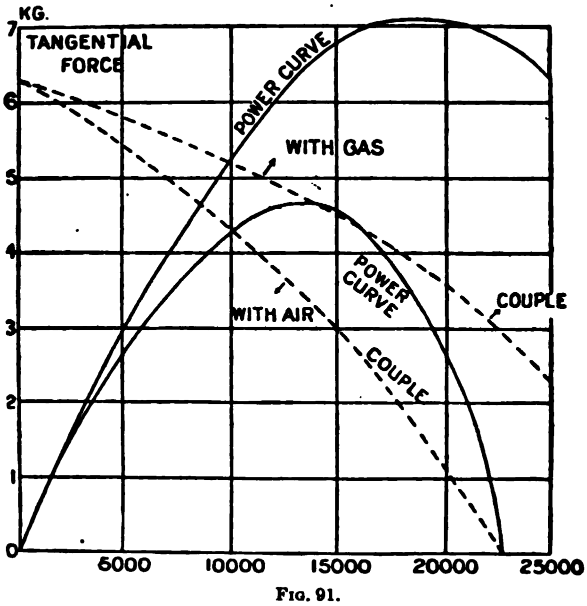

FIG. 91.

The accompanying diagram (Fig. 91) gives the results of practical tests of a Rateau multiple air compressor, as well as a characteristic curve of the small experimental gas turbine of M. Armengaud and Lemale, as communicated to the author by M. Alfred Barbezat, who has been associated with the late M. Réné Armengaud in much of his work.

Experiments with the large turbine and compressor have shown the operative practicability of the machine, but the consumption of petrol (1200 to 1300 grammes per horse-power hour) being too high for industrial purposes. Experiments which we are not yet at liberty to publish, however, indicate that the fuel consumption will be very materially lowered.

The following data concerning gas turbines furnished by the Société des Turbomoteurs to the Creusot Works for use in submarine torpedoes, show the extent to which the practical development of the gas turbine has already attained:

Power	120 horse-power
Speed	1500 revolutions per minute
Expansion ratio	8.4 to 1.4 atmospheres (1:6)
Weight of turbine	73.16 kilogrammes (162 lb.)
Weight of petrol	1.55 kilogrammes (3.4 lb.)
Weight of water	11.00 kilogrammes (24.2)
Weight of air and reservoir, 32 + 177 = 209 kilogrammes (627.6 lb.)	

During the past year there has been built in Paris, by M. Karavodine, an explosion gas turbine developing about 2 horse-power, and operating with regularity and success; and from recent tests of this machine by M. Alfred Barbezat we are able to give some quantitative data concerning its performance.

The Karavodine explosion gas turbine tested by M. Barbezat was provided with four explosion chambers, the products of the explosions being directed through four separate nozzles upon a single turbine wheel. This wheel was of the De Laval type, about 6 inches in diameter (150 centimetres), carried upon a flexible shaft, and fitted with a Prony brake.

The general construction of the explosion chambers is shown in the illustration. The body of the chamber *B* is composed of cast iron and provided with a water jacket *A*, which does not extend all the way to the top, thus permitting the portion nearest the discharge nozzle to become heated. At the lower end there is provided an opening *C* for the entrance of the fuel, either gas or hydrocarbon vapor;

also, an opposite opening *D* for the entrance of air. These openings are both provided with throttle valves, not shown in the illustration, by means of which the proportions of air and gas may be regulated. At *E* is an electric ignition plug, and at *F* is a plate steel valve, opening inward, and held to its seat by a spiral spring *G*, its lift being regulated by a set-screw *H*. The discharge nozzle is shown at *I*, and also a portion of the perimeter of the turbine wheel.

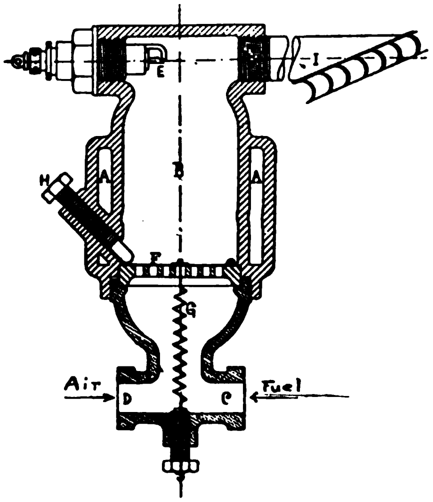

FIG. 92.—Combustion chamber of the Karavodine turbine.

In starting the machine the air opening *D* is closed by its throttle valve and a blast of air is blown through *C*, the explosive mixture being ignited by a spark at *E*. After the first explosion the air entrance *D* is opened and a sort of pulsometer action follows, thus: After each explosion there follows a depression, or partial vacuum, which acts to draw air and hydrocarbon vapor or gas into the chamber *B*, lifting the valve *F*. This mixture is instantly ignited by the spark at *E*, and another explosion follows, to be again followed by another suction, and so on indefinitely.

After a short time the upper part of the chamber *B* becomes so hot that the igniter *E* may be shut off, the charge being exploded by the heat of the chamber. The nozzle *I* is made rather long, and it is found that the friction against the walls and the inertia of its contents prevent any material negative or back suction through it, so that the chamber *B* is filled at every stroke almost entirely from the air and gas openings below.

When the tension on the spring *G* and the lift of the valve *F* are both carefully adjusted this simple device will run for hours, without miss or interruption, the explosions following each other so closely as to make practically a continuous discharge upon the turbine wheel.

In order to investigate the action and pressure in this explosion chamber, a special form of recording gauge was made. The pressure in the chamber acted upon a thin steel diaphragm, of which the deflections actuated a small mirror, throwing a beam of light upon a rapidly-moving, sensitive film. The result was the production of a curve of the sine type, in which the ordinates represent pressure and the abscissæ show time.

In the diagram shown in the illustration the solid curved line is made up from the average of a number of diagrams, while the dotted line shows the one which deviated most widely from the mean. During the period *A E* there was a partial vacuum in the chamber, and the mixed charge was drawn in. From *E* to *D* the pressure of the explosion occurred, and the contents of the chamber were discharged upon the wheel. The ignition began at *B*, and the force of the explosion reached its maximum at *C*, while the period *A B* includes the inertia action of the gases. The diagram shows that a complete oscillation required 0.026 part of a second, corresponding to between 38 and 39 explosions per second. The mean pressure *A F* in the diagram was 1.139 kilogrammes per square centimetre (absolute), or about 16¼ pounds per square inch, the maximum force of the explosion being 1.345 kilogrammes, or about 19 pounds per square inch. The lowest suction pressure was 0.890 kilogramme, or 12.6 pounds absolute, thus giving a negative pressure of

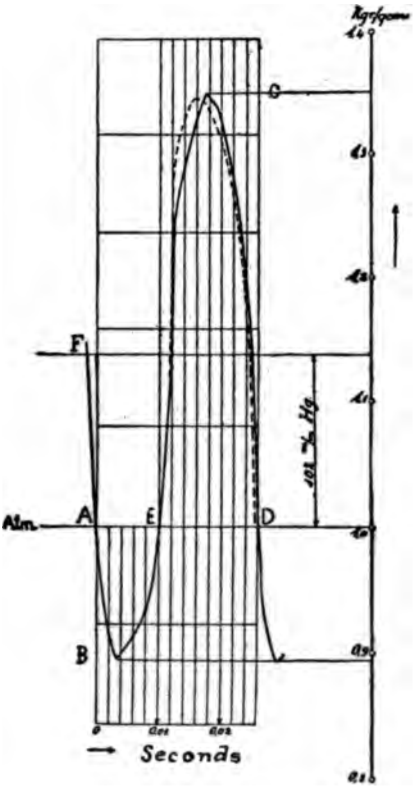

FIG. 93.—Diagram of the explosion turbine.

about 2 pounds to draw the charge in, and a discharge pressure of between 4 and 5 pounds on the wheel.

In the machine tested by M. Barbezat the volume of one chamber was 230 cubic centimetres. Each nozzle was 3 metres long and 16 millimetres bore, slightly curved at the end to conform to the shape of the wheel. The wheel itself was 150 millimetres in diameter, or 5.9 inches, and made 10,000 revolutions per minute, corresponding to a perimeter velocity of 78.5 metres, or about 258 feet per second.

At the same time the above diagrams were taken the amount of air drawn into the four chambers was measured by a meter, and the quantity of gasoline consumed was measured, while the power developed was determined by the Prony brake. The data and results were as follows:

Air consumed per hour, 62.5 cubic metres = 80 kg.

Gasoline, 6.5 litres = 4.7 kg.

Length of brake arm, 46.4 centimetres.

Weight on brake, 248 grammes.

Speed, 10,000 revolutions per minute.

From these figures the brake power works out 1.6 horse-power, and as the wheel and journal friction was determined at 0.5 horse-power, the actual indicated power was 2.1 horse-power. This gives a fuel consumption of 2.24 kilogrammes of gasoline per horse-power hour, which is very fair for such a small machine, being only about one-third greater than that of the old Lenoir gas engine.

In considering the availability of such a machine there are a number of considerations other than the mere fuel consumption. The continuous turning effort is often most desirable, and when it is considered that the wheel of this machine was less than 6 inches in diameter, the possibilities of such an apparatus may become evident. The absence of a compressor and corresponding reduction in weight and size give such a machine marked advantages over the combustion turbine, in which the compressor is much larger than the turbine itself, and even if the fuel consumption is as high as indicated above.

CHAPTER VII.

GENERAL CONCLUSIONS.

In the previous chapters there has been shown broadly the mathematical and thermodynamical principles upon which the possibilities of the construction of a practicable gas turbine may be based, together with some account of the success which has attended the design and operation of actual machines. Much remains to be done before the gas turbine can be expected to enter the market in competition with existing gas engines of the reciprocating type, but there are many active and energetic minds at work upon this portion of the problem, and commercial results may soon be expected to follow.

So far as predictions may be made at this stage of the question, it seems as if the most immediate results are to be anticipated from the so-called "mixed" turbine; the type in which the injection of water for cooling purposes causes the machine to partake of the combined nature of the gas and the steam turbine. This is especially true of the combustion turbine, in which a continuous combustion in a closed chamber provides the gases under pressure to act upon the wheel. The turbine of the explosion type, notwithstanding its low thermal efficiency, appears to have arrived at a practical stage already, and the machine constructed by Karavodine, and tested by Barbezat, has demonstrated that a dry gas turbine of this type is an operative machine already about as efficient as a steam engine of the same capacity.

Apart from the question of thermal efficiency, the development of the gas turbine depends to a large extent upon other properties.

One of the principal difficulties with the reciprocating gas engine lies in the intermittent character of the impelling forces upon the crank shaft, a defect which the multiplication of cylinders in engines designed for automobiles and aeroplanes is intended to remedy as far as practicable with machines of that type. The continuous rotary action of the turbine is such an advantage as to outweigh to a large degree its present lack of fuel economy. In like manner the high rotative speed lends itself to a corresponding reduction in weight per unit of power, a matter which closely concerns the development of mechanical flight. In this matter, as in the case of submarine propulsion, fuel economy is a secondary consideration. The late Professor Langley, in speaking of the engine of his flying machine, is reported to have said that it might burn gold if necessary, so long as it fulfilled all the other requirements of the problem.

The development of the rotary air compressor has an important bearing upon the success of the combustion turbine, and the work of Rateau in this respect has shown what may be accomplished by concentration upon such a question. The analysis of M. Sekutowicz shows the advantages of a high degree of compression, and the high efficiency of the Diesel motor is well known to have resulted largely from the high compressions employed in that most economical heat engine.

What is needed for the further development of the gas turbine, then, is the experimental determination of the data which mathematical analysis has shown to be lacking; data concerning the behavior of gases in diverging nozzles, concerning the action of highly heated gases upon the resistance of materials of construction, data concerning the velocity of efflux from nozzles, data upon the practicability of maintaining extremely high rotating velocities in practical work.

Here is ample work for the engineering laboratories of technical schools; work which can be conducted with existing equipment, and which would form valuable contributions to knowledge, while at the same time providing most fruitful examples for instruction in the very department of engineering in which future progress is to be expected, the subject of the manufacture of power and its utilization to the greatest advantage.

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